ENERGY
MANAGEMENT
SERIES

14

FOR INDUSTRY
COMMERCE
AND INSTITUTIONS

Compressors and Turbines

Energy, Mines and Resources Canada
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PREFACE

Much has been learned about the art and science of managing energy during the past decade. Today, energy management is a seriously applied discipline within the management process of most successful companies.

Initially, in the early 1970's, energy conservation programs were established to alleviate threatened shortages and Canada's dependency on off-shore oil supplies. However, dramatic price increases quickly added a new meaning to the term "energy conservation" — reduce energy costs!

Many industrial, commercial and institutional organizations met the challenge and reduced energy costs by up to 50%. Improved energy use efficiency was achieved by such steps as employee awareness programs, improved maintenance procedures, by simply eliminating waste, as well as by undertaking projects to upgrade or improve facilities and equipment.

In order to obtain additional energy savings at this juncture a greater knowledge and understanding of technical theory and its application is required in addition to energy efficiency equipment itself.

At the request of the Canadian Industry Program for Energy Conservation, the Commercial and Institutional Task Force Program and related trade associations, the Industrial Energy Division of the Department of Energy, Mines and Resources Canada, has prepared a series of energy management and technical manuals.

The purpose of these manuals is to help managers and operating personnel recognize energy management opportunities within their organizations. They provide the practitioner with mathematical equations, general information on proven techniques and technology, together with examples on how to save energy.

For further information concerning the manuals listed below or regarding material used at seminars/workshops including actual case studies, please write to:

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INTRODUCTION

Before the harnessing of power through systems using steam and electricity, natural convection and gravity were used to move air and liquids. The development of powered devices began with the application of power systems to elementary devices such as bellows for compressing air and buckets for moving water.

The availability of low cost power systems allowed devices and systems to become larger and more relied upon. Consumption of energy by the driving devices was overshadowed by the remarkable benefits, and little attention was given to management of the energy input. In recent years the cost of all forms of energy has rapidly increased. This cost burden must be reduced at every opportunity.

Because some types of compressors or turbines are used in many Industrial, Commercial and Institutional facilities, the effective utilization of these devices is an important part of the management of the facility. Energy saved through implementation of *Energy Management Opportunities* can represent a considerable cost saving.

Purpose

The purpose of this module can be summarized by the following.

- Introduce the subject of compressors and turbines as used in the Industrial, Commercial and Institutional sectors.
- Define methods of determining the approximate energy consumption.
- Provide an awareness of potential energy and cost savings available through the implementation of Energy Management Opportunities.
- Provide a series of *Worksheets* to establish a standard method of calculating energy and cost savings for the noted Energy Management Opportunities.

It must be understood that this is not a design manual, but a presentation of information and calculation tools which can be used to identify Energy Management Opportunities, and to estimate potential energy and cost savings.

Contents

The module is divided into separate sections for compressors and turbines. Each section is divided into the following subsections.

- *Fundamentals* describes the basic operating principles and provides simplified equations for estimating the energy requirements of the device. Diagrams illustrate the principles, and worked examples demonstrate the applications of equations.
- *Equipment/Systems* describes the devices and discusses their characteristics with respect to energy consumption.
- A series of *Energy Management Opportunities* provides a suggested list of topics for consideration. Fundamental equations are used in worksheets to produce sample calculations of energy saving, cost saving and simple payback.

Reference material for the sections is in *Appendices*. These include a glossary, tables, common conversions, worksheets and specific details for energy calculations pertaining to *Electric Motor Drives and Alternators*.

Energy Audit Methods

Energy Management Opportunities exist where compressors or turbines are used. Many of these opportunities are recognizable during a *walk through audit* of the facility. This audit is usually more meaningful if a "fresh pair of eyes", generally familiar with energy management, is involved.

Typical opportunities noted during a walk through audit might include an overheated compressor room, or a gas turbine with an air inlet pressure substantially below atmospheric pressure. Alert management, operating staff and good maintenance procedures can, with a little effort, reduce energy use thereby saving money.

Not all items noted during a walk through audit are easy to analyze. For example, it may be observed that warm water from a compressor cooling system is discharged to a floor drain. The immediate reaction is that this water should be directed to a process that could use the heat energy. This leads to the following questions.

- How much water flow is required to cool the compressor and yet keep the water as hot as possible?
- Is the discharge water contaminated?
- Can the water be used elsewhere in the facility?
- Will the recovered energy pay for the modifications?

This example requires a *diagnostic audit* to mathematically establish the potential use of the heat, the reduction in energy consumption and the potential cost saving. With the cost saving plus the estimated cost to supply and install the modified system, simple payback calculations can establish the financial viability of the opportunity.

The implementation of Energy Management Opportunities can be divided into three categories.

- *Housekeeping* refers to an energy management action that is *repeated on a regular basis and never less than once a year.* Examples include the replacement of compressor and turbine inlet filters, and bearing lubrication.
- *Low cost* refers to an energy management action that is *done once and for which the cost is not considered great*. Examples include the relocation of air intakes and control of compressor cooling water flow.
- *Retrofit* refers to an energy management action that is *is done once and for which the cost is significant.* Examples include heat recovery from compressor cooling water and heat recovery from a turbine exhaust.

The division between low cost and retrofit is normally a function of the size, type, and financial policy of the organization.

SECTION 1 — COMPRESSORS

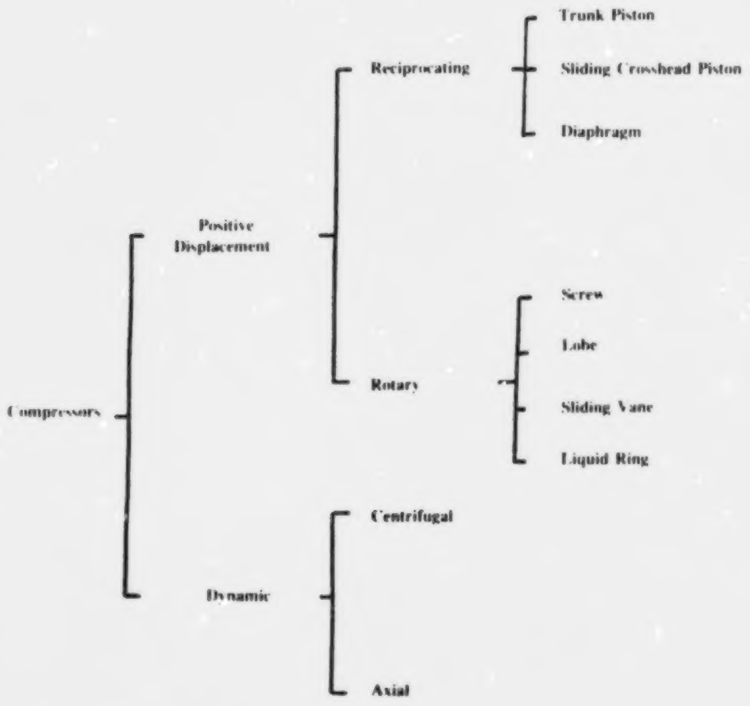
FUNDAMENTALS

SAVE
ENERGY

Compressors are mechanical devices which draw in air and discharge it at a higher pressure, usually into a piping system or tank. These machines can be used to compress room air into a high pressure distribution system or to draw the air from a tank and discharge it into the atmosphere thereby causing a vacuum in the tank. Positive displacement compressors use pistons or rotors to compress the gas, and dynamic compressors use impellers or blades for compression.

Compressor Types

Figure 1 provides a classification of the various types of compressors commonly found in Industrial, Commercial or Institutional facilities. The fundamentals of compressor operation are discussed under the two major classifications, *positive displacement* and *dynamic*.

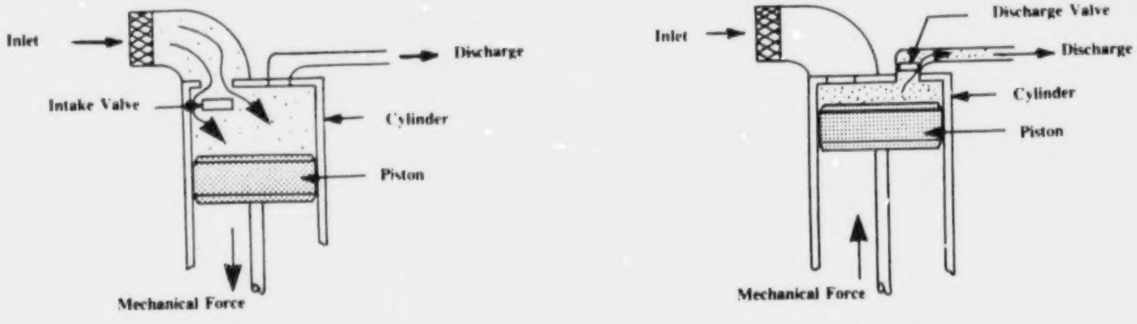

Compressors
Positive Displacement
Dynamic
Reciprocating
Rotary
Centrifugal
Axial
Trunk Piston
Sliding Crosshead Piston
Diaphragm
Screw
Lobe
Sliding Vane
Liquid Ring

Classification of Compressors
Figure 1

Positive displacement compressors may use *reciprocating* or *rotary* action. A reciprocating compressor produces an intermittent discharge flow. It is normally used in combination with a receiver to absorb the resulting pulsation effect and provide a stable discharge pressure. A rotary compressor produces a uniform discharge flow and is often connected directly to a piping system.

The operation of a *positive displacement* compressor can be explained by observing a piston type (Figure 2). Movement of the piston downward (Figure 2a) draws the intake valve open and pulls air into the cavity between the top of the cylinder and the piston. As the piston travels toward the top of the cylinder the intake valve is pushed closed (Figure 2b) and the air is compressed, or squeezed, into.the smaller space. The pressure of the air increases as the volume is reduced. When the pressure in the space above the piston exceeds the pressure in the discharge line, the discharge valve opens and the compressed air moves into the line.

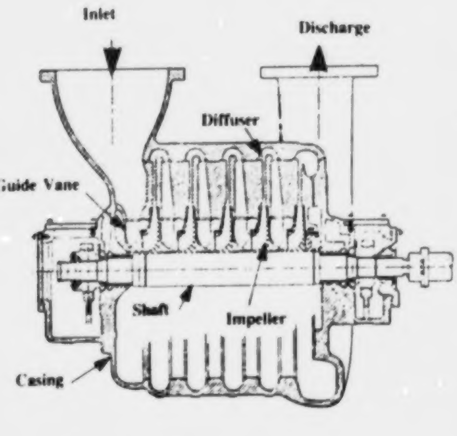
The operation of a *dynamic* compressor can be explained by observing a centrifugal compressor (Figure 3). The inlet air is fed through guide vanes to the first stage of radial blades on the impeller. Rotation of the impeller causes the air to be thrown to the periphery by centrifugal force. Before being guided to the inlet of the next impeller stage the air passes through a diffuser where the kinetic energy of motion is converted to pressure. The process is repeated through successive impeller stages to the compressor discharge. The pressure increase at each impeller stage is determined by the amount of velocity change and the density of the air.



(a) Intake

(b) Compression and Discharge

Positive Displacement Compressor
Figure 2



Centrifugal Compressor
Figure 3

Compressor Operation

Joules Law states that the internal energy of a gas depends only on its temperature and is independent of the volume it occupies. Therefore the energy expended to compress a gas to a higher pressure is represented entirely by an increase in its temperature. Because of this effect, most compressors employ some sort of heat exchange device to remove excess heat from the discharge air.

Gas Laws

Although the actions differ, all compressors obey the gas laws. During changes of conditions such as compression, expansion and heating, air obeys the various laws of thermodynamics and the fundamental gas laws. These laws are combined in the *general gas law* equation which closely approximates the behaviour of air. For changing conditions in a given mass of air, the equation may be written as follows.

$$\frac{P_1 \times V_1}{T_1} = \frac{P_2 \times V_2}{T_2}$$

Where, P_1, P_2 = initial and revised pressures [kPa(absolute)]

V_1, V_2 = initial and revised volumes (L)

T_1, T_2 = initial and revised temperatures (K)

kPa(absolute) = kPa(gauge) + 101.325

Kelvin units (K) = °C + 273.15

The equation can be adapted to convert a volume of air at a measured pressure and temperature to the equivalent volume of air at standard conditions of 20°C (293.15 K) and 101.325 kPa(absolute), called *free air volume.*

$$\text{Equivalent free air volume} = \frac{P \times V \times 293.15}{T \times 101.325}$$

Where, P = measured pressure [kPa(absolute)]

V = measured volume (L)

T = measured temperature (K)

293.15 = standard air temperature (K)

101.325 = standard atmospheric pressure [kPa(absolute)]

Equivalent free air volume is expressed in litres (L).

The general gas law equation can also be stated in terms of volumetric flow rate as follows.

$$\frac{P_1 \times f_{a1}}{T_1} = \frac{P_2 \times f_{a2}}{T_2}$$

Where, f_{a1}, f_{a2} = initial and revised air flow rates (L/s)

This equation can be adapted to convert an air flow to equivalent *free air flow rate* (f_{as}) in units of litres per second (L/s).

$$f_{as} = \frac{P_1 \text{ x } f_{al} \text{ x } 293.15}{T_1 \text{ x } 101.325}$$

Compressor Performance Measurement

The performance of a compressor is determined by calculations based on measured air conditions at the inlet and outlet, and measured flow rate at the outlet. Manufacturer's ratings for compressors are normally stated as equivalent *free air flow rate*, or *standard air flow rate*, at various discharge pressures based on free air intake.

The compressed air flow rate in a pipe can be determined by installing a precalibrated flow measuring device such as an orifice plate, flow nozzle or positive displacement type flow meter. Air and gas flow measuring devices and their characteristics are discussed further in Measuring, Metering and Monitoring, Module 15. The measured flow rate at a particular pressure can be converted to equivalent free air flow rate using the gas law equation.

The flow rate from a positive displacement compressor can be determined by measuring the time required to produce a predetermined pressure rise in a closed pressure vessel, or receiver, of known volume. By measuring the initial and final pressures and temperatures in the receiver, the gas law equation can be applied to calculate the equivalent volume of free air delivered during the measured time period and thus the average free air flow rate. For example, a reciprocating type compressor with a water cooled heat exchanger on the discharge air operated 120 seconds to raise the pressure in a 275 litre receiver from 651 kPa(absolute) (P_1) to 790 kPa(absolute) (P_2). The initial temperature (T_1) was measured to be 22°C (295.15 K) and the final temperature (T_2) was measured to be 29°C (302.15 K).

$$\text{Initial equivalent free air volume} = \frac{P_1 \text{ x } V_1 \text{ x } 293.15}{T_1 \text{ x } 101.325}$$

$$= \frac{651 \text{ x } 275 \text{ x } 293.15}{295.15 \text{ x } 101.325}$$

$$= 1755 \text{ L}$$

Similarly, the equivalent free air volume at the final conditions can be calculated.

$$\text{Final equivalent free air volume} = \frac{P_2 \text{ x } V_2 \text{ x } 293.15}{T_2 \text{ x } 101.325}$$

$$= \frac{790 \text{ x } 275 \text{ x } 293.15}{302.15 \text{ x } 101.325}$$

$$= 2080 \text{ L}$$

The average equivalent free air flow rate can then be calculated.

$$\text{Equivalent free air flow rate} = \frac{\text{Final equivalent free air volume} - \text{Initial equivalent free air volume}}{\text{Time}}$$

$$= \frac{2080 - 1755}{120}$$

$$= 2.71 \text{ L/s}$$

Inlet Conditions

The temperature and pressure of the air at the compressor inlet have a direct effect on compressor performance. As the temperature decreases, the density of air increases. Therefore, the *inlet air to a compressor should be as cool as possible.* With higher density air entering the compressor, a greater quantity of air is compressed for a given inlet volume. Applying the general gas law equation with the inlet pressure constant ($P_1 = P_2$), the effect can be expressed in the following manner.

$$f_{a2} = \frac{f_{a1} \times T_2}{T_1}$$

Consider the effect of compressing an initial flow rate of 1 L/s of outside air at 20°C (293.15 K) versus plant ambient air at 40°C (313.15 K) with a constant displacement compressor which operates intermittently and is controlled by a pressure switch. If the two sources of supply air reach the compressor at the same pressure ($P_1 = P_2$), the effective air flow rate from the plant may be calculated.

$$f_{a2} = \frac{1 \times 313.15}{293.15}$$

$$= 1.07 \text{ L/s}$$

The compressor will require seven per cent greater intake air volume at 40°C than at 20°C inlet air temperature to deliver a particular flow rate of air at the same discharge pressure. This will make it necessary for the compressor to operate more and the energy consumption will thus be increased.

A compressor will use less energy when the restriction to intake air flow rate is reduced. Since a *pressure drop* is required to move the air; intake screens, ductwork and filters should be selected to minimize resistance. When the inlet temperature is constant ($T_1 = T_2$), the effect of inlet pressure drop in such devices can be estimated by the following equation.

$$f_{a2} = \frac{P_1 \times f_{a1}}{P_2}$$

Consider the effect of the following initial and revised inlet pressure conditions on compressor performance.

- Pressure drop across intake piping is 2 kPa for an initial net inlet air pressure (P_1) of 99.325 kPa(absolute).
- Initial air flow rate at inlet (f_{a1}) is 1 L/s.
- Intake piping modified to reduce the pressure drop to zero for a revised net inlet air pressure (P_2) of 101.325 kPa(absolute).

$$\text{Revised air flow rate at inlet, } f_{a2} = \frac{99.325 \times 1}{101.325 - 0}$$

$$= 0.98 \text{ L/s}$$

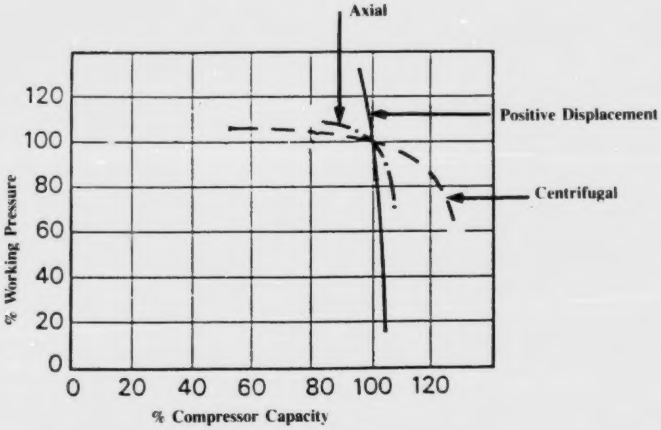
Without the pressure drop through the intake arrangement, two per cent less air volume could be drawn into the compressor to achieve the same discharge rate.

Compressor Performance

Manufacturers' performance tables are the most reliable source of data for compressor operating characteristics. Such performance tables usually list equivalent free air flow rates at certain operating speeds and discharge pressures for the particular manufacturer's standard models. Catalogue tables may only list power inputs in the form of recommended motor sizes. It may be necessary to contact the manufacturer for more detailed data on shaft power inputs and the effects of particular operating conditions. When such data is not available, the power requirement of a particular compressor can be estimated using measured data, the gas laws and the compressor performance equations that are discussed in this section. An example of the use of the developed equations under the heading "Energy Analysis of a Compressor" follows on page 15.

Compressor Performance Characteristics

The capacity of a positive displacement compressor is almost independent of the working pressure but is slightly affected by internal leakage and the volume of gas which remains in the compressor on each compression cycle, or *clearance volume*. The capacity of a dynamic compressor varies with the working pressure in a different relationship for each type and configuration. Figure 4 illustrates typical constant speed performance characteristics for positive displacement, centrifugal, and axial compressors.



Typical Compressor Performance Characteristics
Figure 4

Compressor Efficiency

The power input to a compressor must perform the work of compression and overcome machinery losses. Machinery losses are caused by internal friction, internal gearing, interstage piping and valves. When a compressor is driven directly by an electric motor with or without a coupling, the *power input to the compressor shaft* (Wc_i) can be taken as the motor shaft power output (Wm_o). On a compressor with a power transmission system such as a belt drive, the power input to the compressor shaft is the output of the driving device less the drive loss. The drive loss can be converted to drive efficiency as follows.

$$Ef_d = 1 - \frac{\text{Drive Loss}}{100}$$

Where, Ef_d = drive efficiency (decimal)

Drive loss is expressed in per cent (Table 1)

100 = conversion from per cent to decimal.

Table 1 displays curves that can be used to estimate the drive loss for most fixed ratio power transmission devices if the value is not known. Transmission devices having a ratio of compressor to motor speed greater than 1 will have losses above the average curve, and transmission devices having a ratio of compressor to motor speed of less than 1 will have losses below the average curve.

Appendix E explains the determination of electric motor shaft power output (Wm_o), and the turbine section of this module explains the shaft power output for steam and gas turbines (Wt_o). The power input to a compressor shaft from an electric motor drive system can be calculated from the following equation.

$$Wc_i = Wm_o \times Ef_d$$

Where, Wc_i = power input to the compressor shaft (kW)

Wm_o = motor shaft power output (Appendix E) (kW)

Ef_d = drive efficiency (decimal)

Similarly, for a turbine driven system, the power input to the compressor shaft can be calculated from the following equation.

$$Wc_i = Wt_o \times Ef_d$$

Where, Wt_o = turbine shaft power output (kW)

The power required to compress air with no machinery loss is called the *ideal compression power*. The *compressor efficiency* is the ratio of the ideal compression power to the power input to the compressor shaft.

$$Ef_c = \frac{Wc}{Wc_i} \times 100$$

Where, Ef_c = compressor efficiency (%)

Wc = ideal compression power (kW)

100 = conversion from decimal to per cent

Compressor Power

The following equation can be used to calculate the *ideal compression power*.

$$Wc = 0.00433 \times P_i \times f_{as} \times N \times \left[\left(\frac{P_d}{P_i}\right)^{\frac{0.231}{N}} - 1\right]$$

Where, P_i = inlet pressure [kPa(absolute)]

f_{as} = equivalent free air flow rate (L/s)

N = number of stages

P_d = discharge pressure [kPa(absolute)]

0.00433 = combination of the effect of specific heat of air and conversion of units

0.231 = a function of the specific heat of air

1 = equation constant

The change in the ideal compression power caused by an incremental change in the inlet air temperature to a compressor can be expressed in the following form.

$$Wc_2 = Wc_1 \times (1 + [0.00341 \times (T_{i2} - T_{i1})])$$

Where, Wc_1, Wc_2 = initial and revised ideal compression powers (kW)

T_{i1}, T_{i2} = initial and revised inlet temperatures (K)

1 = equation constant

0.00341 = rate of change per K

Similarly, the change in ideal compression power caused by an incremental change in the inlet air pressure can be expressed as follows.

$$Wc_2 = \frac{Wc_1 \times P_{i1} \times F_{i1}}{P_{i2} \times F_{i2}}$$

Where, P_{i1}, P_{i2} = initial and revised inlet pressures [kPa(absolute)]

F_{i1}, F_{i2} = factors of initial and revised inlet pressures

$$\text{and, } F_{i1} = \left(\frac{P_d}{P_{i1}}\right)^{\frac{0.231}{N}} - 1$$

$$\text{and, } F_{i2} = \left(\frac{P_d}{P_{i2}}\right)^{\frac{0.231}{N}} - 1$$

By substituting for Wc_2 and Wc_1, the previous equations can also be used to calculate revised values for the following.

Wc_{i1}, Wc_{i2} = initial and revised power inputs to the compressor shaft (kW)

Wm_{o1}, Wm_{o2} = initial and revised motor shaft power outputs (kW)

Wm_{i1}, Wm_{i2} = initial and revised motor power inputs (kW)

Capacity Control Methods

The method of controlling a compressor's capacity when the full output is not required can have an important effect on its energy consumption. Methods that control the speed or running time of the compressor result in energy consumption that is approximately proportional to the output. Methods that reduce the internal capacity of the compressor, called *unloading devices*, have various effects on the energy consumption depending on the type of device and the characteristics of the compressor.

Particular compressor control devices and their characteristics are discussed in the Equipment section of this module.

Operation Costs and Savings

The cost of energy to operate a compressor that is driven by an electric motor can be determined by the electrical measurements and calculations described in Appendix E.

When the electrical power input to the motor has been determined, the following equation can be used to calculate the annual energy cost ($/yr).

$$\text{Annual energy cost} = Wm_i \times Ce \times h$$

Where, Wm_i = motor power input (kW)

Ce = unit electrical energy cost ($/kWh)

h = operation time (h/yr)

The total cost of energy to operate compressors that are equipped with *unloading devices* can be estimated by the following equation.

$$\text{Annual energy cost} = [(Wm_{iL} \times h_L) + (Wm_{iu} \times h_u)] \times Ce$$

Where, Wm_{iL}, Wm_{iu} = motor power inputs − loaded and unloaded (kW)

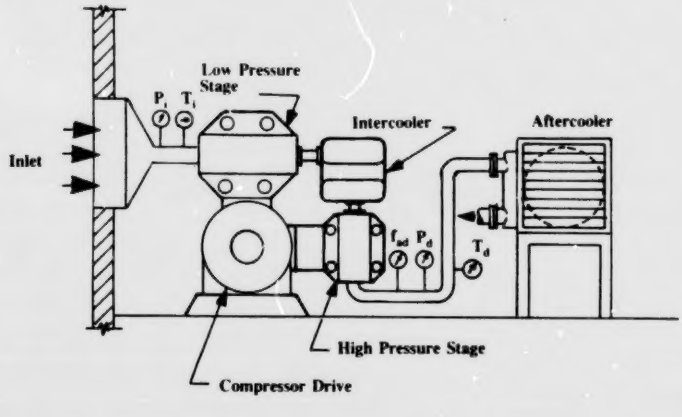
h_L, h_u = operation times − loaded and unloaded (h/yr)

This simplified equation is suitable for single step unloading systems or compressors that unload in a manner that can be evaluated as a single step.

Compressors equipped with *variable speed controllers* can be evaluated to an acceptable degree of accuracy using the average power requirement for the total operation period.

Energy Available for Recovery

Because the energy expended by a compressor is represented entirely by an increase in the air temperature, all compressors use some form of cooling to reduce the temperature of the discharge air. Small compressors may use heat dissipating fins on the compressor casing. In multiple stage compressors the air is normally cooled between stages in an intercooler (Figure 5), and in many installations the air is also cooled after the final stage by an aftercooler.



Arrangement Of Compressor Components
Figure 5

Cooling the compressed air reduces the specific volume which may allow the use of smaller pipe and/or reduce the friction losses. *Most of the energy expended to compress the air is recoverable* by utilizing the intercooler and aftercooler as heat sources for other useful purposes.

Table 2 lists the components of an air compressor that have potential for energy recovery. The values listed represent the maximum portion of input energy that is recoverable as heat from the cooling air and water. The values are referred to as the *heat exchange efficiencies*. The maximum heat recovery from a compressor component cooled by air, gas or liquid can be estimated by the following equation.

$$Q_{pr} = 3.6 \times Wc_i \times Ef_e$$

Where, Q_{pr} = potential heat recovery (MJ/h)

Ef_e = total portion of input energy released (total of heat exhange efficiences from Table 2) (decimal)

3.6 = conversion from kW to MJ/h.

For example, an air cooled compressor with a shaft power input of 38 kW and equipped with an air cooled intercooler is examined. Table 2 indicates that a combined air cooled compressor and intercooler can release 0.50 of the shaft input energy. The potential for heat recovery can be calculated.

$$Q_{pr} = 3.6 \times 38 \times 0.50$$

$$= 68.4 \text{ MJ/h}$$

The value of recovered energy becomes a saving when the energy is used to displace an energy source that has a cost. Savings are affected by the unit cost of the energy source. A list of common sources of energy and their higher heating values is compiled in Appendix C. By recovering the heat energy represented by Q_{pr}, additional saving may be realized owing to the reduction in load on the facility cooling system.

The annual energy cost saving ($/yr) resulting from heat recovery that is used to replace *fuel fired heat* can be approximated by the following equation.

$$\text{Annual fuel cost saving} = \frac{Q_{pr} \times Cf \times h}{HHV \times Ef_b}$$

Where, Cf = unit fuel cost ($/unit)

h = operation time (h/yr)

HHV = fuel higher heating value (MJ/unit) (Appendix C)

Ef_b = efficiency of the fuel burning device (decimal) (if unknown, use 0.75)

The annual saving ($/yr) resulting from a reduction in the consumption of purchased *steam* may be determined as follows.

$$\text{Annual steam cost saving} = \frac{Q_{pr} \times Cs \times h}{2.2}$$

Where, Cs = unit steam cost ($/kg)

2.2 = approximate usable heat energy of steam (MJ/kg)

When recovered heat is used to replace *electric heat*, the annual saving ($/yr) may be estimated as follows.

$$\text{Annual electrical energy cost saving} = \frac{Q_{pr} \times Ce \times h}{3.6}$$

Where, Ce = unit electrical energy cost ($/kWh)

3.6 = conversion from kWh to MJ

Methods of calculating the value of electrical energy saved by reducing the energy consumption of *electric motors* are presented in Appendix E.

In some instances the heat liberated from compressor components places a load on a building cooling system. The following equation can be used to calculate the annual energy cost saving ($/yr) owing to a cooling load reduction.

$$\text{Annual cooling energy cost saving} = \frac{Q_{pr} \times Cc \times Sc \times h}{1000}$$

Where, Cc = unit cost of energy for cooling ($/unit)

Sc = energy consumption for cooling (units/GJ)

1000 = conversion of GJ to MJ

If the energy consumption for cooling (Sc) is unknown, use 80 kWh/GJ for electrically driven systems and 0.45 kg/GJ for steam driven systems.

Energy Analysis of a Compressor

The following analysis uses the fundamentals and equations presented in the previous discussions.

A two stage water cooled air compressor (Figure 5) in a central heating plant was rated at 500 L/s of air with a normal working pressure of 700 kPa(gauge). It was equipped with a filtered air inlet piped from outside the plant, and an air cooled intercooler. The compressor was electrically driven through a multiple V belt drive.

The heating plant electrician collected the following motor nameplate data.

Rated voltage, V_r	575 volts
Rated full load current, I_r	177 amps
Number of phases	3
Power rating	147 kW
Rated full load power factor, $p.f._r$	0.91

The electrician took measurements and obtained the following average values.

Measured voltage, V	579 volts
Measured current, I	178 amps
Measured power factor, p.f.	0.91

Records maintained for the various measurement points on the compressor, as indicated in Figure 5, resulted in the following average values.

Inlet pressure, P_i	−4.6 kPa(gauge) [96.725 kPa(absolute)]
Discharge pressure, P_d	691 kPa(gauge) [792.325 kPa(absolute)]
Air flow rate at discharge pressure, f_{ad}	81.9 L/s
Inlet temperature, T_i	10.3°C (283.45 K)
Discharge temperature, T_d	99.7°C (372.85 K)

General plant records indicated that the compressor operates for 2000 hours per year and electricity costs an average of $0.05 per kWh.

The discharge air flow rate was converted to equivalent free air flow rate (f_{as}) using the general gas law equation.

$$f_{as} = \frac{P_d \times f_{ad} \times 293.15}{T_d \times 101.325}$$

$$= \frac{792.325 \times 81.9 \times 293.15}{372.85 \times 101.325}$$

$$= 503 \text{ L/s}$$

Ideal compression power was determined from the following equation.

$$Wc = 0.00433 \times P_i \times f_{as} \times N \times \left[\left(\frac{P_d}{P_i}\right)^{\frac{0.231}{N}} - 1\right]$$

$$= 0.00433 \times 96.725 \times 503 \times 2 \times \left[\left(\frac{792.325}{96.725}\right)^{\frac{0.231}{2}} - 1\right]$$

$$= 116 \text{ kW}$$

The actual power required by the compressor was calculated from the collected data and the electrical measurements. The equations used in these calculations are explained in Appendix E.

The load on the electric motor was determined from the following equation.

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{178 \times 579 \times 0.91}{177 \times 575 \times 0.91}$$

$$= 1.01$$

The load ratio was an acceptable operating condition because certain motor designs can operate at 15 per cent overload.

An electric motor with a 1.01 load ratio would have an efficiency of 0.92 as obtained from Figure E-1. The motor power output was then determined.

$$Wm_o = \frac{V \times I \times Y \times p.f. \times Ef_m}{1000}$$

$$= \frac{579 \times 178 \times 1.73 \times 0.91 \times 0.92}{1000}$$

$$= 149 \text{ kW}$$

From Table 1 the V belt drive loss for a 149 kW motor power output was read as 2.8 per cent. The resulting drive efficiency was then calculated.

$$Ef_d = 1 - \frac{\text{Drive Loss}}{100}$$

$$= 1 - \frac{2.8}{100}$$

$$= 0.97$$

The power input to the compressor shaft was then calculated.

$$Wc_i = Wm_o \times Ef_d$$

$= 149 \times 0.97$

$= 145 \text{ kW}$

The compressor efficiency was determined from the calculated values for ideal compression power and power input to the compressor shaft.

$$Ef_c = \frac{Wc}{Wc_i} \times 100$$

$$= \frac{116}{145} \times 100$$

$$= 80\%$$

If the unit had been direct coupled the estimated drive loss of 2.8 per cent would have been saved.

As indicated by Table 2, a typical air cooled intercooler can release approximately 45 per cent of the compressor energy input. Similarly, Table 2 indicates the heat energy released by the compressor to be 4 per cent of the energy input.

Total portion of input energy released, $Ef_e = 0.45 + 0.04$

$= 0.49$

The heat recovered from the intercooler and compressor could be used to preheat domestic hot water and replace steam consumption.

Potential heat recovery, $Q_{pr} = 3.6 \times Wc_i \times Ef_e$

$= 3.6 \times 145 \times 0.49$

$= 256 \text{ MJ/h}$

With the unit cost of steam production (Cs) at \$22/1000kg, the annual cost saving can be calculated.

$$\text{Annual steam cost saving} = \frac{Q_{pr} \times Cs \times h \times Ef_b}{2.2}$$

$$= \frac{256 \times 0.022 \times 2000 \times 0.75}{2.2}$$

$$= \$3{,}840/\text{yr}$$

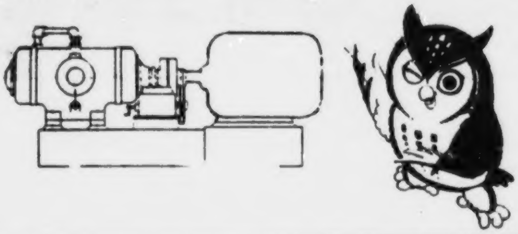
Summary

Electric motors, turbines and combustion engines convert particular forms of energy to shaft power which is used to produce the compressed air. Alert design, operations and maintenance personnel, with an awareness of energy management, can increase the effective use of energy for compressors.

The heat energy carried away from the compressor in the compressed air is lost unless appropriate measures are implemented to put it to use.

Compressors can be analyzed in detail to estimate the energy input, energy transfer, losses, costs and potential savings. Worksheets 14-C1 through 14-C7 demonstrate the data and calculations necessary for the analysis of compressors.

EQUIPMENT SYSTEMS



As listed in Figure 1 and discussed in the Fundamentals section, the types of compressors addressed in this module are positive displacement and dynamic types. Positive displacement compressors are subdivided into reciprocating and rotary types. Dynamic compressors are subdivided into centrifugal and axial flow types.

Positive Displacement Compressors

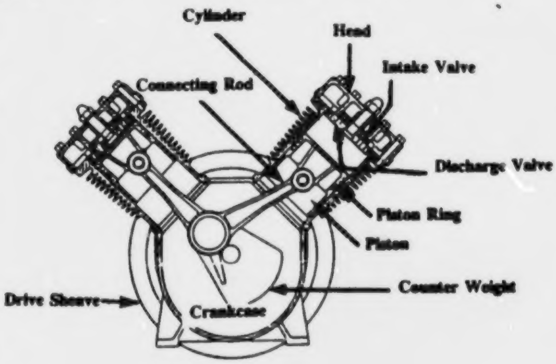
Positive displacement compressors compress air or gas by reducing the volume it occupies in a cylinder or rotor. Their capacities are not significantly affected by the working pressure.

Reciprocating Compressors

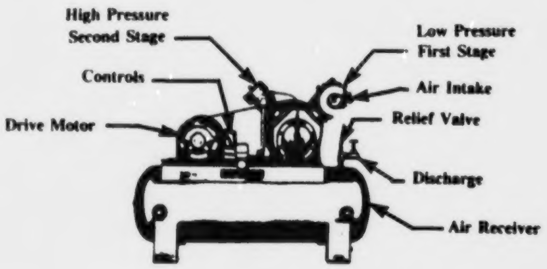
Reciprocating compressors are the most common type of positive displacement compressor. Table 3 lists the common characteristics of such units. Smaller units are used where modest quantities of air are required for garages, manufacturing plants, shops, pressurizing tanks, instrument control and air powered tools. Larger sizes are used for industrial plant air systems, operating equipment and machines, boiler soot blowing and instrument air. Reciprocating compressors are also used for compressing gases such as carbon dioxide, hydrogen, nitrogen and refrigerants.

Reciprocating compressors are widely used because they are simple to operate and maintain, low cost and compact. They operate at low speeds, are easily regulated, and have good efficiency over the capacity range. A disadvantage is the generation of internal heat caused by friction. The reciprocation action causes vibration which may necessitate more robust foundations than for other types.

The design of *trunk piston compressors* (Figure 6) limits them to single-acting and lubricated applications. The piston is connected to the crankshaft by a connecting rod, and the bottom of the cylinder is open to the crankcase. Since the bottom of the cylinder cannot be sealed, compression is restricted to the top of the piston, making the unit *single-acting*. Lubricating oil in the crankcase splashes on the exposed cylinder walls or is sprayed on the walls. Because of this *lubricated* configuration, a portion of the lubricant is carried out with the discharged air as a contaminant. Single-stage units may have two or more cylinders, and are usually air cooled. Two-stage machines tend to be larger, and are cooled by either air or water. Higher capacity units are usually water cooled. Most facilities that have a modest demand for compressed air use single-acting compressors. As shown in Figure 7, small compressors may be mounted directly on an air receiving tank and form a package system complete with piping, controls, relief valve and motor.



Trunk Piston Compressor
Figure 6



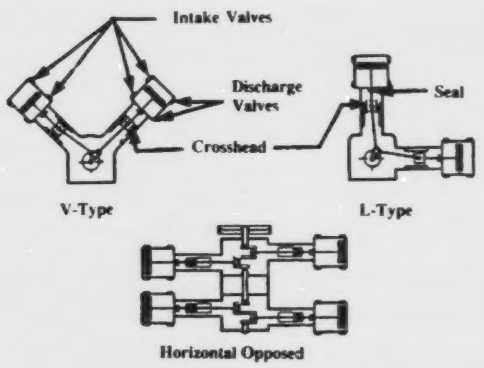
Small Compressor Package
Figure 7

The *sliding crosshead piston compressor* (Figure 8) has a rigid rod connecting the piston to the crosshead. The crosshead is attached to the crank shaft by a connecting rod. The rigid rod moves through a seal allowing compression on both sides of the piston (double-acting). The seal also isolates the cylinder from the crankcase. Compressors of this type are always double-acting but may be single or multistage. Owing to the isolated piston these machines are available with oil lubrication or oil-free. The oil-free design uses special self-lubricating materials or coatings on the piston, cylinder walls and piston rings. Oil-free air is often required for special industrial applications, instrument air, and institutional medical air. These compressors are available in vertical, vertical V, L-type and horizontal configurations, and for various capacities and pressure ratings. Multicylinder horizontal units can have discharge pressures up to 60 MPa(gauge). Air or water may be used for compressor cooling but the larger units are normally water cooled.

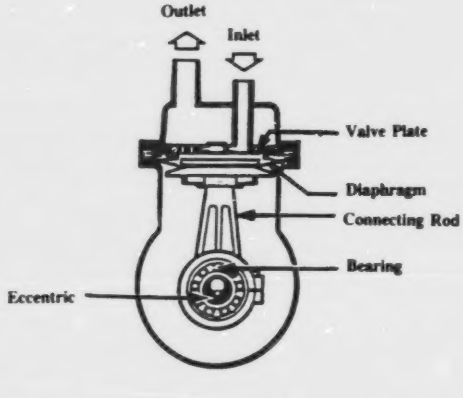
A modified version of the sliding crosshead compressor uses the labyrinth design. Instead of sealing rings there are concentric grooves on the piston and in the cylinder walls. Although leakage past the piston is high, the air held in the grooves separates the surfaces resulting in minimal piston friction. This design provides contaminant free discharge air but at reduced efficiency.

A *diaphragm compressor* uses a flexible membrane, or diaphragm, to perform the same function as a piston. A common design (Figure 9) has the connecting rod pinned to a block, or piston, which is attached to the diaphragm. Another type has the piston separated from the diaphragm by a chamber filled with fluid by which the diaphragm is moved hydraulically. Because of the isolation effect of the diaphragm, these compressors provide oil-free air.

Diaphragm compressors are usually inexpensive, light, compact units of simple construction, and can be easily maintained. Because they have low internal friction, cool discharge air is delivered at low noise level. Diaphragm compressors can withstand impurities in the air since the air does not come in contact with the piston.



Sliding Crosshead Piston Compressors
Figure 8



Diaphragm Compressor
Figure 9

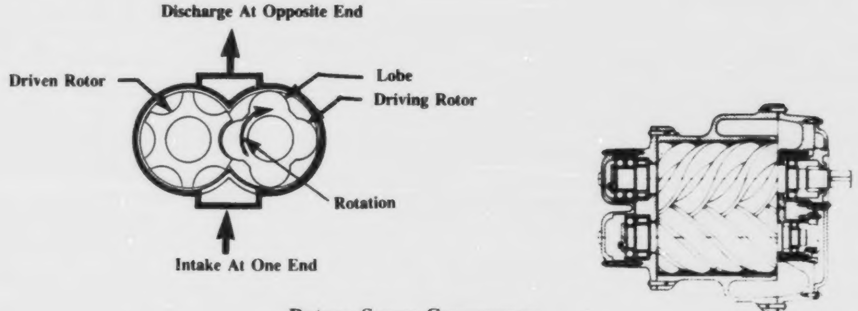
Rotary Compressors

Rotary compressors, sometimes called rotaries, are now being used in applications once dominated by reciprocating machines. *Positive displacement* rotaries can be direct coupled to driving motors and operate at high speeds. Their overall weight, size and capital cost are usually less than those of comparable reciprocating machines. This is possible since rotaries have no intake or outlet valves and no unbalanced mechanical forces. These machines have good full load but poor part load efficiency. The low part load efficiency is caused by leakage between mating surfaces. Common capacities, pressures and drive sizes are listed in Table 4.

The *rotary screw compressor* (Figure 10) has a driving screw with four spiral starts and lobes. The driven rotor has six starts and cavities into which the lobes fit. Inlet air is trapped in the moving cavities between the rotors and the casing, and compression is accomplished by the termination of the cavities at the discharge port. Some designs use gears driving both rotors so the rotors do not touch. This reduces the heat of friction within the compressor and allows oil-free operation. These compressors are commonly used in the capacity range from 150 to 1000 L/s. They are cooled by air or water. The advantages of the rotary screw machines include good reliability, low overall capital and operating costs, good full load efficiency and tolerance of impure air.

These machines are used for the same purposes as the medium sized reciprocating units.

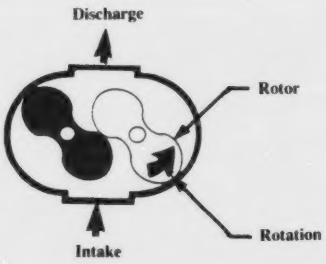
- General plant air.
- Instrumentation and controls.
- Chemical and oil refining processes.
- Vacuum drying.
- Pneumatic conveying.
- Mine and tunnel ventilation.
- Snow making.
- Sootblowing.



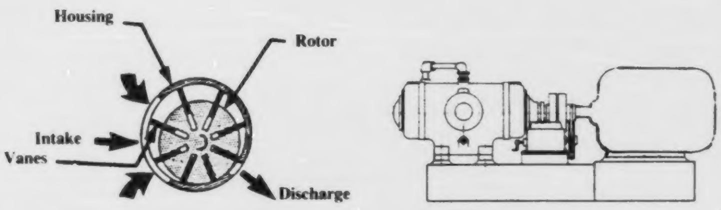
Rotary Screw Compressor
Figure 10

Lobe type air compressors (Figure 11), sometimes called blowers, are used in low pressure applications. The gear driven rotors transfer the air from the inlet to the outlet port where the air is compressed to the discharge line pressure. The advantages of lobe compressors are high volumetric efficiency, low mechanical friction and low vibration. The positive displacement characteristics of these compressors makes them suitable for measuring flow. Because the rotors are gear driven and do not contact, neither lubrication nor cooling is required, allowing oil-free operation. This type of compressor is used for materials handling, aeration, and as an air pressure booster on an internal combustion engine.

The *sliding vane compressor* (Figure 12) has a solid rotor mounted eccentrically within a circular housing. As the rotor turns, thin blades or vanes move in and out of slots in the rotor maintaining a seal against the interior of the housing. The eccentric location of the rotor causes air to be drawn into the chamber, compressed and then discharged by the changing volume of the spaces between the vanes. Cooling is accomplished by using air, water or oil. Oil cooled units impart the heat from circulated lubricating oil to air or water in a heat exchanger. Units below 1 kW may be operated oil-free with self-lubricating carbon or fiberous vanes. Advantages are that these machines may be direct driven and have no unbalanced forces (low vibration). Disadvantages include low efficiency, high lubrication oil consumption and poor single stage capacity control. They are used for general air service and instrumentation air.



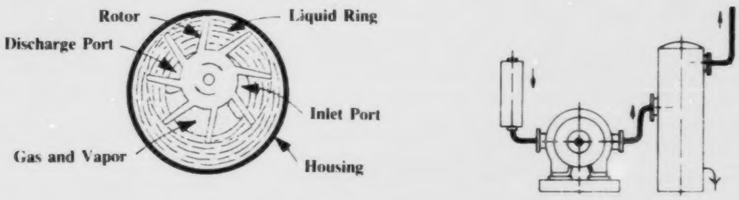
Lobe Compressor
Figure 11



Sliding Vane Compressor
Figure 12

The *liquid ring compressor* (Figure 13) has a multi-blade rotor mounted eccentrically within a circular housing that is partially filled with liquid. As the rotor spins, the liquid in the housing is thrown outwards by centrifugal force, forming a ring of liquid. Because of the eccentric position of this ring of liquid, the spaces in the rotor fill with liquid and then empty as the blades rotate. The continuous cycle of filling and emptying creates a pumping effect causing suction and discharge of the air or gas.

Liquid ring compressors are used with central vacuum and compressed air systems in hospitals, laboratories, schools and some industrial establishments. As vacuum pumps, the machines have the ability to accept water, other liquids and soft waste materials that may be drawn into the system. As compressors the machines have the ability to clean dust and bacteria from the air without adding oil or other contaminants. The liquid is recirculated outside the housing providing a means of cooling the air during compression. Liquid ring compressors have low efficiency, are limited to low discharge pressure (Table 4) and must have the accessories to handle the sealing liquid.



Liquid Ring Compressor
Figure 13

Dynamic Compressors

Dynamic compressors are used mainly in large industrial applications and for specialized applications such as turbine inlet air compression. Their common characteristics are given in Table 5.

Centrifugal Compressors

A centrifugal compressor (Figure 14) causes the air to travel radially from the impellers and pass through diffusers between stages prior to discharge. The air can be effectively cooled between stages by cooling the housing, resulting in near ideal compression stages. Except in the largest sizes, the overall efficiency of centrifugal compressors is less than that of positive displacement machines because of the significant energy loss in the diffusers. Centrifugal compressors provide a stable discharge pressure with wide variations in air flow rate. Interstage cooling of centrifugal compressors is normally accomplished by water circulated through the casing. Units that discharge at less than 400 kPa(gauge) do not normally require cooling.

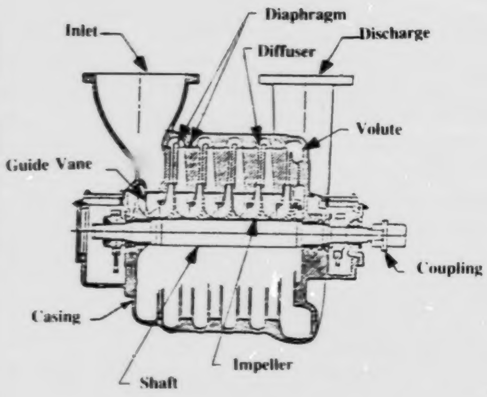
Centrifugal compressors operate at high speeds with most commercial units running at approximately 20 000 revolutions per minute (rpm). In the aircraft and space industries speeds of 100 000 rpm can occur.

The advantages of centrifugal compressors become significant at flow rates in excess of 1200 L/s. The major advantages are large capacity, low vibration, compact construction, oil-free air discharge and the self-limiting capacity of the units. The major disadvantages include the necessity of a speed increaser (unless turbine driven), close running clearances, and high maintenance cost.

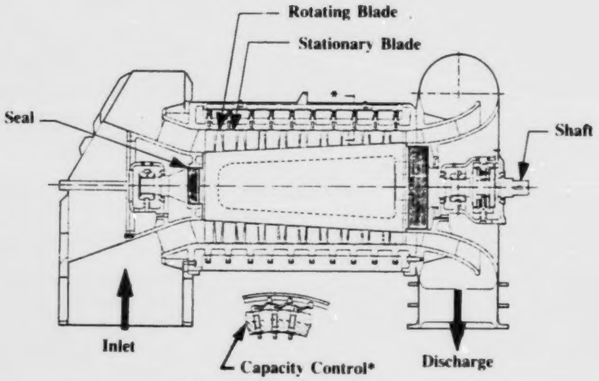
Axial Compressors

Air movement through an axial compressor (Figure 15) is parallel to the shaft, and occurs through alternating rows of rotating and stationary blades. Pressure rise through an individual stage or rotating blade is restricted because of the difficulty of cooling the air within the casing. In order to discharge air at similar conditions to that of a centrifugal compressor, the speed of an axial unit must be 25 per cent higher.

The major advantages of the axial compressor is the lower capital and operating costs for capacities above 65 000 L/s and oil free discharge air. The main disadvantage is that, for stable operation, the air flow demand must be relatively constant and close to the operating range. These machines are used for supplying combustion air to gas turbines, and for blast furnace air in the steel industry.



Centrifugal Compressor
Figure 14



Axial Compressor
Figure 15

Air Receivers

An air receiver is useful in plants where air demand is not constant, and where the air compressor is not used at maximum capacity at all times. An air receiver allows the compressors to be sized for average plant load. Peak demand loads are supplied by the stored air. The air receiver prevents short cycle loading and unloading of the compressor, enabling the compressor to run at higher efficiency for longer periods of time. Pressure surges caused by load changes, or pulsations in air lines caused by reciprocating compressors, are dampened out by the receiver. More than one receiver is sometimes used where there are long runs of piping, or where there is large or irregular air use. Heat loss from the receiver results in partial cooling of the compressed air, and collection of water condensate and oil droplets. The packaged compressor and air receiver in Figure 7 allows the use of an inexpensive fixed capacity compressor on a system with fluctuating air consumption.

The installation of an appropriately sized air receiver in a multiple compressor installation may permit a reduction in the number of compressors operated at any one time for a saving in peak power demand cost in addition to the operating cost saving.

Compressor Control

One of three basic methods is used to control a compressor's capacity when its full output is not required.

Constant speed control has the compressor run continuously while varying the capacity by one of several *compressor unloading systems*. Most large compressors use this method of control because the large electric drive motors cannot withstand numerous starts.

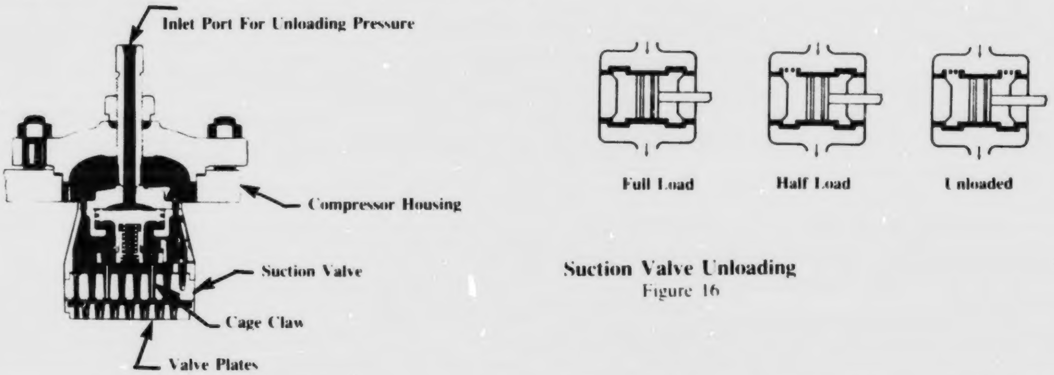
Start-stop control usually uses a pressure sensing switch to start and stop the compressor. This method is used almost universally on small compressors that deliver air to a receiver. Some larger compressors are controlled by start-stop devices when their capacity exceeds twice the air demand.

Dual control combines the two previous methods to allow selection of the appropriate method depending on the operating conditions. Selection may be manual or automated.

Compressor Unloading Systems

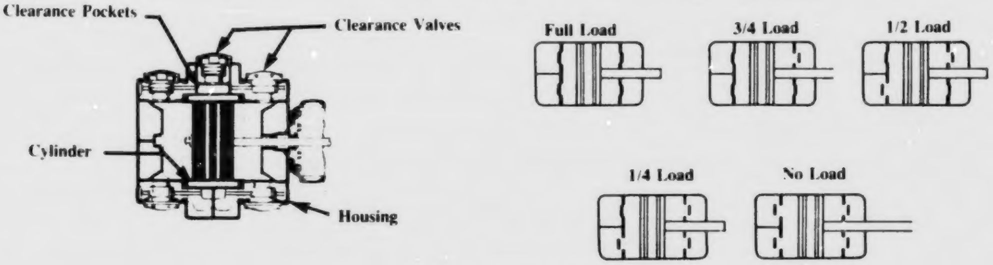
Compressor unloading systems reduce the output of a compressor by the use of devices that reduce the internal volumetric capacity of the compressor. Such systems are combined with motor control to provide both volume control and energy conservation.

Suction valve unloading (Figure 16) utilizes a claw cage that pushes open the plates of the suction valve when the demand for air is reduced. A stage of a reciprocating compressor with two suction valves can be progressively unloaded to provide three step control: full load, half load and no load. With this type of unloading the energy consumption is roughly proportional to the reduced load.


Inlet Port For Unloading Pressure
Compressor Housing
Suction Valve
Cage Claw
Valve Plates
Full Load
Half Load
Unloaded

Suction Valve Unloading
Figure 16

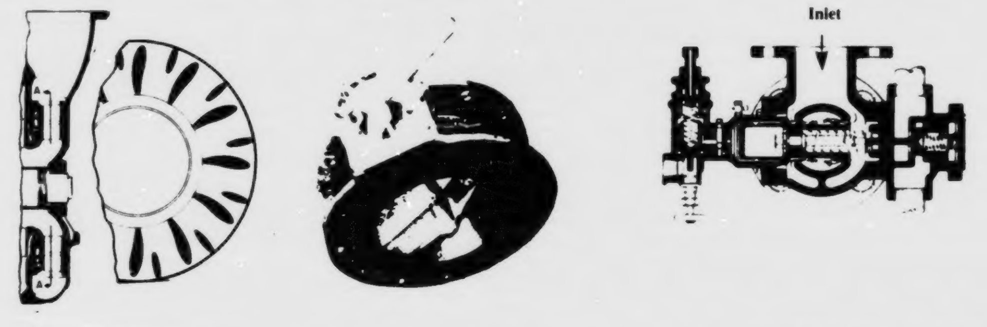
Clearance pocket unloading (Figure 17) utilizes single or multiple valves interconnecting the cylinders to internal clearance pockets within the housing. Opening the valves increases the volume of the cylinder and lowers the volumetric efficiency and the compressor's ability to deliver air. This type of unloading method can have numerous steps but the energy consumption remains relatively high during unloaded operation.


Clearance Pockets
Clearance Valves
Cylinder
Housing
Full Load
3/4 Load
1/2 Load
1/4 Load
No Load

Clearance Pocket Unloading
Figure 17

Inlet throttling unloading utilizes a continuously variable valve or inlet guide vanes (depending on compressor type) on the compressor inlet (Figure 18). As the valve closes, the flow rate of air to the compressor is reduced. This method increases the compressor pressure ratio resulting in relatively high unloaded energy consumption.

By-pass control and *blow-off control* unloading systems allow the compressor to continue to deliver air but the excess air is either by-passed back to the inlet or vented at the discharge. These systems result in no reduction in energy consumption under part load operation. Not only is the energy consumption high but, with the by-pass system, more heat energy must also be removed from the air.



Adjustable Inlet Guide Vanes **Hand Operated Butterfly Valve** **Inlet Unloader For Sliding Vane Compressor**

Compressor Unloading Devices
Figure 18

Intercoolers And Aftercoolers

Intercoolers are used between stages of a multiple stage compressor to lower the air temperature and thereby improve the performance of subsequent stages. *Aftercoolers* lower the temperature of the compressed air at the compressor outlet to reduce the amount of moisture in the air. They also increase the air density, which reduces the volumetric flow rate, thereby reducing the pressure drop in the system piping.

Intercoolers and aftercoolers are available as air cooled or water cooled units. Aftercoolers that use refrigeration are also available for systems that require very dry compressed air. Figure 5 shows a typical arrangement of a two stage compressor involving an air cooled intercooler and an air cooled aftercooler.

Related System Components

The performance of some common system and process components have an important impact on compressor performance. The performance characteristics of filters and air intake screens can vary significantly during their service life owing to obstruction by dust and other airborne particles. Such components should therefore be considered when evaluating a compressor system.

ENERGY MANAGEMENT OPPORTUNITIES

Energy Management Opportunities is a term that represents the ways that energy can be used wisely to save money. A number of Energy Management Opportunities, subdivided into Housekeeping, Low Cost, and Retrofit categories are outlined in this section, with worked examples or text to illustrate the potential energy savings. This is not a complete listing of the opportunities available for compressors. However, it is intended to provide ideas for management, operating, and maintenance personnel to allow them to identify other opportunities that are applicable to a particular facility. Other modules in this series should be considered for Energy Management Opportunities applicable to other types of equipment and systems.

Housekeeping Opportunities

Implemented housekeeping opportunities are energy management actions that are done on a regular basis and never less than once a year. The following are typical Energy Management Opportunities in this category.

1. Check for leaks in the air system.
2. Check the operation of compressed air system coolers and clean heat exchange surfaces regularly.
3. Provide the coolest air practicable to air compressors.
4. Implement a program of inspection and planned maintenance to minimize compressor component failures.
5. Correct abnormal noise or vibration of compressors to ensure smooth and efficient operation.
6. Check and adjust drives regularly to maintain proper belt tension, sheave and coupling alignment.
7. Shut down compressors when compressed air supply is not required.
8. Clean or replace compressor intake air filters on a regular basis to prevent excessive wear of moving components and restriction of air flow.
9. Set controls to operate at lowest possible pressure levels and flow rates.

Housekeeping Worked Examples

The worksheets can be used to estimate the potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets.

1. Check for Leaks in the Air System

Savings in air compression energy result from the correction of leaking shutoff valves, leaking valve stems, leaking pipe fittings, and leaking of damaged piping and hoses. Even though they do not drip, emit odors or present any apparent danger, compressed air leaks should not be ignored. The savings obtained by reducing air system leaks are discussed in Water and Compressed Air Systems, Module 12.

2. Maximize Cooler Performance

Coolers associated with air compressors must be kept clean and provided with an adequate flow of water or air for maximum performance. Cooling air that has not been filtered, or which is drawn through poor filters, can foul cooler passages and fan blades. Contaminated cooling water can foul heat exchanger surfaces and build up deposits on the interior of the circulation piping. The compressed air side of coolers can be fouled by dirt in the air combining with condensed water and oil vapors. To help combat these conditions, the following procedures should be a part of compressor housekeeping.

- Clean or replace filters as recommended by the manufacturer, or when the pressure drop becomes excessive.

- Ensure filters fit correctly and no leakage occurs.
- Run cooling fans and circulating pumps when the compressor is running and test that pressurized cooling water is flowing.
- Remove flow obstructions in air passages.
- Test for contamination of cooling water.
- Check that the unit is operating efficiently by measuring the operation parameters and comparing them to the design conditions.

3. Provide Coolest Air to Compressors

As discussed under Fundamentals, the coolest possible intake air to the compressors results in the least energy for compression.

When the air is taken from inside the plant, close doors to hot areas of the plant, turn off unnecessary lighting around the compressors, reduce building temperatures in the compressor areas and open exterior windows to serve as air intakes when the outside air temperature is safely above freezing.

4. Maintenance Program

A maintenance program for compressors should be tailored to the specific needs of the facility and should include the following items.

- Daily: Monitor compressor sounds, vibration, lubricating oil level and temperature, various seals and connections, leaks, auxiliary equipment, and the reading of installed gauges and meters.
- Monthly: Examine air intake screens and filters, check lubricating oil for contamination, grease appropriate fittings, test unloading system, change oil filters, examine shaft couplings, check tightness of all exposed connections and bolts, and check condition of all auxiliary equipment.
- Annually: Calibrate all instrumentation, conduct a compressor performance test and perform required repairs, thoroughly test all auxiliary equipment, and clean, examine, and lubricate all bearings.

5. Noise and Vibration

Abnormal noise or vibration of a compressor could be caused by one or more factors.

- Worn components such as piston rings, valves, gearing, linkages and impellers.
- Bad bearings and bushings.
- Inadequate lubrication.
- Inadequate cooling.
- Dirty components.

By analyzing the vibration characteristics, it is often possible to identify the source of the problem and schedule corrective action before the efficiency of the compressor is seriously affected.

6. Check Pulleys and Drives

Compressor drives, which include belts and couplings, provide long service when properly designed and maintained. The following maintenance actions should be performed regularly.

- Correct the alignment of sheaves and couplings.
- Correct the tension of belts.
- Provide lubrication as required.
- Replace or repair damaged belts, chains, sheaves, sprockets, clutches, drive keys and couplings.

Proper belt tension must be maintained because loose belts can cause slipping, squealing, low compressor speed and rapid belt wear. Sheaves, bearings, shafts and motor will be hot indicating an energy loss. Belts should be tensioned to manufacturer's recommendations, and must be readjusted for stretching after the first forty-eight hours of operation. Excessively tight belts will reduce compressor and motor bearing life and increase the drive loss. Proper tensioning for various types of belts and chains is described in handbooks and drive catalogues by component manufacturers.

7. Shut Systems Down

A 75 kW electric motor driven air compressor, used in a manufacturing plant, operated for 12 hours per day and 250 days per year. It was determined that the compressor could be periodically shut down by the line foreman since it ran unloaded for a total of one hour per day or 250 hours per year.

With the compressor unloaded, the plant electrician obtained the following information.

Number of phases	3
Measured voltage, V	575 volts
Measured current draw, I	23 amps
Measured power factor, p.f.	0.71

Plant energy power bills were used to determine the average unit electrical energy cost to be $0.05/kWh. Worksheet 14-El was used to determine that shutting the compressor down would save $203 per year.

8. Clean or Replace Intake Filters

Clean and efficient intake air filters are essential to the reliability of a compressor. Dust and other impurities entering the compressor can cause sticking valves, scored cylinders and excessive wear on piston rings, bearings, seals and other mating surfaces.

Dirty filters also restrict the flow of inlet air, which increases the energy required by the compressor. By cleaning or replacing the intake filters on a regular basis, the inlet pressure drop is reduced and energy is saved. See Low Cost Worked Example 2 for a calculation of the potential savings.

9. Reduce Operating Pressure

Since the power required by a compressor is directly proportional to the operating pressure, energy cost can be saved by operating at the lowest pressure that will satisfy the system requirements.

Low Cost Opportunities

Implemented low cost opportunities are energy management actions that are done once and for which the cost is not considered great. The following are typical Energy Management Opportunities in this category.

1. Modify or relocate air intakes to cooler locations.
2. Modify or replace components with units of higher efficiency, such as lower pressure drop filters or larger piping.
3. Install flow control devices on cooling system heat exchangers to provide stable operating temperatures and prevent excess water flow.
4. Install a pressure switch or time clock to shut down the compressors when building operations no longer require compressed air.

Low Cost Worked Examples

The worksheets can be used to estimate potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets.

1. Relocate Air Intakes

A two-stage reciprocating air compressor located in an auto assembly plant had its air intake located in the compressor room. The average ambient air temperature in the compressor room was 25°C (298.15 K). To save energy the compressor intake could be ducted to bring in outdoor air. It was estimated that the intake system would result in an overall pressure drop of 1.35 kPa between outdoors and the compressor intake flange, causing a net revised inlet pressure (P_{i2}) of 99.98 kPa(absolute).

The average yearly outdoor temperature is 12°C (285.15 K). An electrician determined that the 150 kW drive motor was consuming 164 kW of electric power during normal operation. The plant operating records for the compressor contained the following data.

Initial pressure at inlet, P_{i1}	101.30 kPa(absolute)
Discharge pressure, P_d	793 kPa(absolute)
Operation time, h	4000 h/yr
Unit electrical energy cost, Ce	$0.05/kWh

Worksheet 14-C1 was used to calculate the effects of the reduced inlet temperature. The revised motor power input was calculated to be 157 kW and the annual energy cost saving was calculated to be $1,400 per year.

The revised motor power output (Wm_{i2}) from Worksheet 14-C1 was then used to determine (Wm_{i1}) in Worksheet 14-C2 to calculate the incremental effect of the revised inlet pressure. The effect was found to cause an increase in the energy cost of $200 per year.

The net annual energy cost saving was therefore $1,400 — $200 = $1,200 per year.

The estimated cost of installing an intake system to draw cooler air from outdoors is $2,830.

$$\text{Simple payback} = \frac{\$2{,}830}{\$1{,}200}$$

$$= 2.4 \text{ years.}$$

2. Low Pressure Drop Filters

A single stage rotary screw type air compressor rated at 250 L/s at free air conditions had a working pressure of 860 kPa(gauge) [961.325 kPa(absolute)] and a restrictive intake filter with a pressure drop of 3 kPa for a net initial inlet pressure (P_{i1}) of 98.325 kPa(absolute). Through a search of catalogues, the maintenance supervisor selected an acceptable replacement filter rated at 1 kPa resistance for a revised inlet pressure (P_{i2}) of 100.325 kPa(absolute). The electric motor was directly connected to the compressor.

The supervisor obtained the following data at initial operating conditions.

Measured voltage, V	580 volts
Measured current, I	141 amps
Measured power factor, p.f.	0.88
Number of phases	3
Unit electrical energy cost, Ce	$0.05/kWh
Operation time, h	4022 h/yr

Worksheet 14-E1 was used to calculate the initial motor power input to be 125 kW.

Worksheet 14-C2 was used to calculate the annual energy cost saving to be $201 per year.

The cost to install lower pressure drop filters is estimated to be $728.

$$\text{Simple payback} = \frac{\$728}{\$201}$$

$$= 3.6 \text{ years.}$$

3. Reduce Cooling System Water Flow Rates

A compressor room in a process plant contained six water cooled compressors which used a measured cooling water flow rate of 1.2 L/s. Discussions with the compressor manufacturer revealed that, by installing a flow control valve on each compressor cooling line, the flow rate could be reduced by 25 per cent, or 0.3 L/s. The compressor plant operator obtained the following data from existing records.

Unit water cost, Cw	$0.20/m^3$
Operation time, h	3120 h/yr

Using Worksheet 14-C3 the total annual saving is calculated to be $674 per year. The addition of six control valves is estimated to cost $1,500.

$$\text{Simple payback} = \frac{\$1,500}{\$674}$$

$$= 2.2 \text{ years}$$

4. Shut Down Compressors

A 75 kW air compressor normally operated for 3120 hours per year. During lunch and coffee breaks the compressor ran unloaded. The plant engineer determined that a time clock could be installed to turn the compressor off during the break periods for a total reduction in operation time of 390 hours per year.

The plant electrician took the following measurements on the three phase drive motor when the compressor was running unloaded.

Measured voltage, V	582 volts
Measured current, I	22 amps
Measured power factor, p.f.	0.72

Using Worksheet 14-E1 the annual saving by not operating the compressor was calculated to be $310 per year based on a unit electrical energy cost of $0.05 per kWh.

The installation of a time clock and associated electrical wiring were estimated to cost $862.

$$\text{Simple payback} = \frac{\$862}{\$310}$$

$$= 2.8 \text{ years}$$

Retrofit Opportunities

Implemented retrofit opportunities are energy management actions that are done once and for which the cost is significant. Many of the opportunities in this category will require detailed analysis by specialists and are beyond the scope of this module. Worked examples are provided for some of the listed Energy Management Opportunities, while in other cases there is only commentary. The following are typical Energy Management Opportunities in the retrofit category.

1. Install a heat recovery system to extract heat from compressor cooling water for reuse elsewhere in the facility. Examples of such systems include fluid to fluid heat exchangers or hot water coils in air handling units.

2. Install enclosures around compressors to trap and exhaust unwanted hot or moisture laden air directly outdoors. If enclosures are not possible, relocate compressors to isolated areas having individual cooling systems.
3. Where possible, use heated air from compressors to reduce building heating loads. The hot air could be used as air curtains at loading docks and as plant makeup air. Where the hot air cannot be used directly, air to air heat exchangers can be used.
4. Install pressure blowers in place of air compressors when only low pressure air is required.
5. Replace single stage air compressors with higher efficiency two stage compressors.
6. Install air-cooled compressed air aftercoolers in series with water-cooled units to assist the plant heating system and reduce cooling water consumption.
7. Install variable speed control on compressor motor to optimize energy consumption versus compressed air demand.
8. Replace a central air compressor and distribution system with multiple compressors located near the points of use.
9. Install a microprocessor energy management control system.
10. Install an air receiver to permit the air compressor to operate at maximum efficiency under fluctuating loads.

Retrofit Worked Examples

The worksheets can be used to estimate the potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets.

1. Heat Recovery of Cooling Water

A three stage reciprocating type air compressor located in a process plant was driven by an electric motor through a speed reducer. The cooling water was directed to drain after cooling the compressor and attached intercooler. An opportunity existed to use the cooling water heat energy in a heat exchanger to reduce No. 2 fuel oil consumption for process heating.

The plant maintenance staff prepared a tabulation of measured and recorded values for analysis.

Electric motor nameplate power	450 kW
Number of phases	3
Rated voltage, V_r	2300 volts
Rated full load current, I_r	127 amps
Rated full load power factor, $p.f._r$	0.92
Measured voltage, V	2334 volts
Measured current, I	121 amps
Measured power factor, p.f.	0.92
Rated motor efficiency (nameplate), Ef_{mr}	0.95
Unit fuel cost (No. 2 oil), Cf	\$0.40/L
Operation time, h	4480 h/yr

Using Worksheet 14-E1, the motor shaft power output was calculated to be 427 kW. Since the compressor was driven through a gear reducer, the power input to the compressor shaft was calculated on Worksheet 14-C4 to be 417 kW.

Worksheet 14-C5 was used to determine the potential annual fuel cost saving to be $44,537 per year.

The total cost of the retrofit including piping, heat exchanger, electrical equipment, controls and safety devices was estimated to be $19,600.

$$\text{Simple payback} = \frac{\$19{,}600}{\$44{,}537}$$

$$= 0.44 \text{ year (5 months)}$$

2. Enclose Compressors

The heat from an air cooled compressor, including the intercooler and motor, was rejected directly into the building space. Consequently, the building ventilation air had to be cooled for an additional 1200 hours per year in order to offset the heat gain from the compressor. It was recognized that cooling energy costs could be reduced by installing an enclosure over the compressor and providing a controlled circulation of outside air through the enclosure.

The electric motor was connected to the compressor by a V-belt drive. The plant electrician took the following readings and measurements.

Rated voltage, V_r	575 volts
Rated current, I_r	135 amps
Rated power factor, $p.f._r$	0.89
Measured voltage, V	580 volts
Measured current, I	109 amps
Measured power factor, p.f.	0.90
Nameplate shaft power output	112 kW
Number of phases	3
Unit electrical energy cost, Ce	$0.05/kWh
Operation time, h	1200 h/yr

Worksheet 14-E1 was used to calculate the motor shaft power output to be 90.5 kW.

Worksheet 14-C4 was used to calculate the power input to the compressor shaft to be 87.5 kW.

The annual saving in cooling energy was calculated on Worksheet 14-C5 to be $922 per year.

An enclosure around the compressor and an air supply fan were estimated to cost $4,650.

$$\text{Simple payback} = \frac{\$4{,}650}{\$922}$$

$$= 5.0 \text{ years}$$

The cost of operating the supply fan was not considered significant for these calculations. The energy cost for fans is discussed in Fans and Pumps, Module 13 in this series.

3. Compressor Heat Recovery

A compressor room located in a processing plant contained five 75 kW screw type, direct coupled, air-cooled air compressors. Each unit had an air-cooled intercooler. The compressor room was isolated from the rest of the plant, but it was determined that the heated air of the compressor room could be used directly to offset some of the winter heating load in the plant. The plant heating system used No. 6 fuel oil at a cost of $0.22/L.

From Environment Canada data, the average heating season for the location was determined to be 36 weeks. With the compressors operating 12 hours per day and 5 days per week, the annual operation time when heat could have been reclaimed was 2160 hours. Since four of the compressors usually ran continuously at high loads, the heat available for reclaim was estimated from the nameplate ratings.

Using Worksheet 14-C5, the potential annual fuel cost saving was calculated to be $9,793 per year.

The cost of the addition of ductwork, filters, fans and controls was estimated to be $21,700.

$$\text{Simple payback} = \frac{\$21{,}700}{\$9{,}793}$$

$$= 2.2 \text{ years}$$

The actual saving would be reduced slightly by the operation cost of the fan. Fans and Pumps, Module 13 discusses the energy costs for fans.

4. Air Compressor Replaced by Pressure Blower

An industrial process used air at a pressure of 55 kPa(gauge) [156.325 kPa(absolute)]. The air was supplied by a single stage reciprocating compressor through an air receiver tank. The compressor was equipped with internal unloading devices which maintained the air receiver at 225 kPa(gauge) [326.325 kPa(absolute)]. From observations by the operating staff, the compressor operated approximately 6596 hours per year loaded and 1828 hours per year unloaded. Air was drawn from the receiver continuously for 8424 hours per year.

The maintenance superintendent decided to calculate the saving that would result from replacing the compressor with a pressure blower (lobe type). To provide sufficient data for the calculations, the superintendent had the following readings and measurements compiled for the existing system.

Measured voltage-loaded, V_L	584 volts
Measured current-loaded, I_L	419 amps
Number of phases	3
Measured power factor-loaded, $p.f._L$	0.86
Operation time-loaded, h_L	6596 h/yr
Measured voltage-unloaded, V_u	587 volts
Measured current-unloaded, I_u	251 amps
Measured power factor-unloaded, $p.f._u$	0.32
Operation time-unloaded, h_u	1828 h/yr
Measured air flow rate from receiver, f_{al}	1385 L/s
Unit electrical energy cost, Ce	$0.05/kWh
Initial air temperature, T_1	44°C (317.15 K)

Worksheet 14-El was used to determine that the annual cost of energy for the reciprocating compressor was $120,047 while loaded (sheet 1) and $7,495 while unloaded (sheet 2), for a total initial annual energy cost of $127,542 per year.

To assist in the selection of a replacement compressor, Worksheet 14-C6 (sheet 3) was used to determine that the equivalent free air flow rate to the process was 1975 L/s. Catalogue information indicated that an appropriate lobe type air compressor was available with a rating of 1976 L/s free air flow rate and 55 kPa(gauge) discharge pressure. The compressor was listed as requiring 132 kW of shaft input power. Using Worksheet 14-C4 (sheet 4), the required motor shaft power output was calculated to be 136 kW. An electric motor catalogue indicated that the smallest motor of sufficient size is rated at 149 kW.

Worksheet 14-E2 (sheet 5) was used to calculate the electrical power input to the proposed motor to be 148 kW and the revised annual energy cost to be $62,338 per year.

The cost to purchase and install the new compressor and a larger delivery line to the process was estimated to be $244,000.

$$\text{Simple payback} = \frac{\$244{,}000}{\$127{,}542 - \$62{,}338}$$

$$= 3.7 \text{ years.}$$

5. Higher Efficiency Compressors

A single-acting, single stage compressor in a process plant was monitored for several operating days, and the results indicated it ran 4000 hours per year. The feasibility of replacing the compressor with a more efficient unit was considered.

The building maintenance superintendent made a length of inlet duct to fit the compressor intake to assist in taking an air flow measurement. He was able to compile the following readings and measurements.

Measured voltage, V	233 volts
Measured current, I	58 amps
Measured power factor, p.f.	0.86
Number of phases	3
Drive arrangement	V belts
Initial air flow rate, f_{a1}	50 L/s
Initial air pressure (inlet), P_1	0 kPa(gauge) [101.325 kPa(absolute)]
Initial temperature (inlet), T_1	16°C (289.15 K)
Discharge pressure, average	515 kPa(gauge) [616.15 kPa(absolute)]
Air receiver shut-off pressure	620 kPa(gauge) [721.15 kPa(absolute)]
Unit electrical energy cost, Ce	$0.05/kWh

Using Worksheet 14-El, the motor power input was calculated to be 20.1 kW.

To determine the rating of a replacement compressor, the air flow rate into the existing compressor was converted to equivalent free air flow rate on Worksheet 14-C6. A compressor sales firm offered a single-acting two stage unit which could provide the required 51 L/s free air flow rate at the 620 kPa(gauge) shut-off pressure. The compressor was listed as requiring 16.1 kW power input to the compressor shaft at the maximum pressure. It was directly driven by an 18.6 kW electric motor. At the average discharge pressure of 515 kPa(gauge), the compressor was listed as requiring 14.7 kW power input to the compressor shaft.

Using Worksheet 14-E2, the saving by replacing the compressor and drive was determined to be $820 per year. The installed cost of the compressor less the salvage value of the existing unit was estimated to be $4,140.

$$\text{Simple payback} = \frac{\$4{,}140}{\$820}$$

$$= 5.0 \text{ years}$$

6. Addition of Air Cooled Aftercooler

An air compressor was belt-driven by a 75 kW motor and had a water cooled aftercooler. It was speculated that an air cooled aftercooler installed in series with the existing aftercooler could reduce part of the building heating load during the heating season. Records indicated that the compressor would operate for 2000 hours during the heating period. From outdoor temperature records and plant heating load records, it was estimated that approximately 80 per cent of the rejected heat could be effectively used for building heating. The average unit steam cost was $0.022/kg.

Using Worksheet 14-C4, the required power input to the compressor shaft was estimated to be 72.5 kW.

Worksheet 14-C5 was used to determine that the annual steam cost saved by using the aftercooler heat would be $2,340 per year.

The installed cost of the new air cooled aftercooler and ducting was estimated to be $5,300.

$$\text{Simple Payback} = \frac{\$5{,}300}{\$2{,}340}$$

$$= 2.3 \text{ years}$$

If the water cooled aftercooler flow was being piped to a drain and wasted, a further saving would occur with the reduction in water consumption.

7. Variable Speed Drives

An electric motor driven reciprocating air compressor with an unloading device was considered for installation of a variable frequency motor speed control device. Records indicated that the compressor operated for 3072 hours per year fully loaded and 3881 hours per year unloaded. The unit electrical energy cost was determined to be $0.05/kWh.

In order to determine the potential savings, the following measurements were taken.

Compressor rated full load output	200 L/s
Operation time-loaded, h_L	3072 h/yr
Operation time-unloaded, h_u	3881 h/yr
Motor nameplate shaft power output	75 kW
Number of phases	3
Rated voltage, V_r	575 volts
Rated full load current, I_r	91.2 amps
Rated full load power factor, p.f.$_r$	0.91
Measured voltage-loaded, V_L	575 volts

Measured current-loaded, I_L	90.0 amps
Measured power factor-loaded, $p.f._L$	0.90
Measured voltage-unloaded, V_u	580 volts
Measured current-unloaded, I_u	32.7
Measured power factor-unloaded, $p.f._u$	0.71

Worksheet 14-E1 was used to determine that the annual energy cost to operate the compressor with the unloading device was $12,380 while loaded (sheet 1), and $4,521 while unloaded (sheet 2), for a total energy cost of $16,901/yr.

On Worksheet 14-E1 (sheet 1) the motor power output was calculated to be 74.2 kW when the compressor was loaded.

The variable speed controller literature stated that the unit evaluated would have an internal efficiency of 95 per cent. Using Worksheet 14-C7 (sheet 4), the operation cost with the variable speed controller was calculated to be $13,037 per year.

The installed cost of a variable speed controller for the compressor motor was estimated to be $34,300.

$$\text{Simple payback} = \frac{\$34{,}300}{\$16{,}901 - \$13{,}037}$$

$$= 8.9 \text{ years.}$$

8. Replace Central Compressors with Multiple Units

Large central compressed air stations with low efficiency and considerable maintenance should be compared with smaller high efficiency compressors sized for individual loads and located near the point of use. Large central compressors that are sized for the peak load of the entire plant and the highest pressures will not operate at peak efficiency when the full air flow rate is not required. Multiple compressors can operate at the peak efficiency for longer periods of time when sized for individual loads. The individual systems may be cross connected for back-up purposes.

9. Install Microprocessor Compressor Management System

A microprocessor compressor management system can accomplish energy cost savings over and above the savings from individual actions by monitoring and integrating various control functions. The analysis and selection of such equipment should be based on a professional review of the requirements for the particular facility. In a process setting, compressor control is often integrated into a larger control network.

10. Install an Air Receiver

An electric motor driven air compressor that operated in a factory was directly connected to the compressed air distribution system. It ran continuously 365 days per year, 24 hours per day, or 8760 hours per year. Pressure measurements indicated that the compressor often ran when compressed air was not required and overpressured the system.

It was recognized that, by installing an air receiver, the compressor could be shut off during the periods when air was not required. The energy consumed during this time represented a potential cost saving. The unit electrical energy cost was determined to be $0.05/kWh.

Measurements were taken during overpressurization to provide the following data.

Measured voltage, V	230 volts
Measured current, I	21 amps

Measured power factor, p.f.	0.79
Number of phases	3
Unit electrical energy cost, Ce	\$0.05/kWh
Operation time (overpressurized), h	1102 h/yr

Using Worksheet 14-El, the annual cost of energy to operate the compressor when it was overpressurizing the system was determined to be \$364 per year. This would be the amount of the annual saving if an air receiver were to be installed.

The installed cost of an air receiver was estimated to be \$1,420.

$$\text{Simple payback} = \frac{\$1{,}420}{\$364}$$

$$= 3.9 \text{ years.}$$

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE #7 Date: 86/08/19

Location: COMPRESSORS - HOUSEKEEPING By: MBE

Rated voltage, V_r N/A volts

Rated full load current, I_r N/A amps

Measured voltage, V 575 volts

Measured current, I 23 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate shaft power output N/A kW

Rated full load power factor, $p.f._r$ N/A (decimal)

Measured power factor, p.f. 0.71 (decimal)

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h (REDUCTION) 250 h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 23 \times 1.73 \times 0.71}{1000}$$

= 16.2 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = N/A kW

Annual energy ~~cost~~ saving = Wm_i x h x Ce = 16.2 x 250 x 0.05 = $ 203 /yr

Compressor Inlet Air Temperature

Worksheet 14-C1
(Page 1 of 1)

Company: WORKED EXAMPLE #1 (SHT. 1) Date: 86/08/19
Location: COMPRESSORS - LOW COST By: MBE

Initial inlet temperature, T_{i1} 298.15 K

Revised inlet temperature, T_{i2} 285.15 K

Initial motor power input, Wm_{i1} 164 /kW

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h 4000 h/yr

Revised motor power input,

$Wm_{i2} = Wm_{i1} \times (1 + [0.00341 \times (T_{i2} - T_{i1})])$

$= 164 \times (1 + [0.00341 \times (285.15 - 298.15)])$

$=$ 157 kW

Annual energy ~~cost~~ saving = $(Wm_{i1} - Wm_{i2}) \times h \times Ce$

$= (164 - 157) \times 4000 \times 0.05$

= $ 1,400 /yr

Compressor Inlet Pressure

Worksheet 14-C2

(Page 1 of 2)

Company: WORKED EXAMPLE #1 (SHT. 2) Date: 86/08/19

Location: COMPRESSORS - LOW COST By: MBE

Data

Number of stages, N	2	
Discharge pressure, P_d	793	kPa(absolute)
Initial inlet pressure, P_{i1}	101.30	kPa(absolute)
Revised inlet pressure, P_{i2}	99.98	kPa(absolute)
Initial motor power input, Wm_{i1} (FROM WORKSHEET 14-C1)	157	kW
Unit electrical energy cost, Ce	$ 0.05	/kWh
Operation time, h	4000	h/yr

Factor of initial inlet pressure,

$$F_{i1} = \left(\frac{P_d}{P_{i1}}\right)^{\frac{0.231}{N}} - 1$$

$$= \left(\frac{793}{101.30}\right)^{\frac{0.231}{2}} - 1$$

$$= 0.268$$

Compressor Inlet Pressure

Worksheet 14-C2
Page 2 of 2

Company: WORKED EXAMPLE #1 (SHT. 2) Date: 86/08/19

Location: COMPRESSORS - LOW COST By: MBE

Factor of revised inlet pressure,

$$F_{i2} = \left(\frac{P_d}{P_{i2}}\right)^{\frac{0.231}{N}} - 1$$

$$= \left(\frac{793}{99.98}\right)^{\frac{0.231}{2}} - 1$$

$$= 0.27$$

Revised motor power input, $Wm_{i2} = Wm_{i1} \times \frac{P_{i1} \times F_{i1}}{P_{i2} \times F_{i2}}$

$$= 157 \times \frac{101.30 \times 0.268}{99.98 \times 0.270}$$

$$= 158 \text{ kW}$$

Annual energy cost ~~saving~~ = $(Wm_{i1} - Wm_{i2}) \times h \times Ce$

$$= (157 - 158) \times 4000 \times \$0.05$$

$$= \$(-)\ 200 \text{ /yr}$$

(EXTRA COST)

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: Worked Example #2 (Sht. 1) Date: 86/08/19

Location: Compressors - Low Cost By: MBE

Rated voltage, V_r	N/A	volts
Rated full load current, I_r	N/A	amps
Measured voltage, V	580	volts
Measured current, I	141	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	N/A	(decimal)
Measured power factor, p.f.	0.88	(decimal)
Unit electrical energy cost, Ce	$ N/A	/kWh
Operation time, h	N/A	h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 141 \times 1.73 \times 0.88}{1000}$$

= 125 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = N/A kW

Annual energy cost saving = $Wm_i \times h \times Ce$ = x x = $ N/A /yr

Compressor Inlet Pressure

Worksheet 14-C2

(Page 1 of 2)

Company: WORKED EXAMPLE #2 (SHT. 2) Date: 86/08/19

Location: COMPRESSORS - LOW COST By: MBE

Number of stages, N 1

Discharge pressure, P_d 961.325 kPa(absolute)

Initial inlet pressure, P_{i1} 98.325 kPa(absolute)

Revised inlet pressure, P_{i2} 100.325 kPa(absolute)

Initial motor power input, Wm_{i1} (FROM WORKSHEET 14-E1) 125 kW

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h 4022 h/yr

Factor of initial inlet pressure,

$$F_{i1} = \left(\frac{P_d}{P_{i1}}\right)^{\frac{0.231}{N}} - 1$$

$$= \left(\frac{961.325}{98.325}\right)^{\frac{0.231}{1}} - 1$$

$$= \underline{\quad 0.693 \quad}$$

Compressor Inlet Pressure
Worksheet 14-C2
Page 2 of 2

Company: WORKED EXAMPLE #2 (SHT. 3) Date: 86/08/19

Location: COMPRESSORS - LOW COST By: MBE

Factor of revised inlet pressure,

$$F_{i2} = \left(\frac{P_d}{P_{i2}}\right)^{\frac{0.231}{N}} - 1$$

$$= \left(\frac{961.325}{100.325}\right)^{\frac{0.231}{1}} -$$

$$= \underline{\quad 0.685 \quad}$$

Revised motor power input, $Wm_{i2} = Wm_{i1} \times \frac{P_{i1} \times F_{i1}}{P_{i2} \times F_{i2}}$

$$= 125 \times \frac{98.325 \times 0.693}{100.325 \times 0.685}$$

$$= \underline{\quad 124 \quad} \text{ kW}$$

Annual energy ~~cost~~ saving = $(Wm_{i1} - Wm_{i2}) \times h \times Ce$

$$= (125 - 124) \times 4022 \times \$0.05$$

$$= \$\underline{\quad 201 \quad}/\text{yr}$$

Cooling Water Saving

Worksheet 14-C3

(Page 1 of 1)

Company: WORKED EXAMPLE #3 Date: 86/08/19

Location: COMPRESSORS - LOW COST By: MBE

Cooling water flow rate, f_w	0.3 L/s
Unit water cost, Cw	\$ 0.20 /m³
Operation time, h	3120 h/yr

Annual water cost saving = f_w x Cw x h x 3.6

= 0.3 x 0.20 x 3120 x 3.6

= \$ 674 /yr

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE # 4 Date: 86/08/19

Location: COMPRESSORS - LOW COST By: MBE

Rated voltage, V_r N/A volts

Rated full load current, I_r N/A amps

Measured voltage, V 582 volts

Measured current, I 22 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate shaft power output N/A kW

Rated full load power factor, $p.f._r$ N/A (decimal)

Measured power factor, p.f. 0.72 (decimal)

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h (REDUCTION) 390 h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{582 \times 22 \times 1.73 \times 0.72}{1000}$$

= 15.9 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m =$ x = N/A kW

Annual energy ~~cost~~ saving = $Wm_i \times h \times Ce$ = 15.9 x 390 x 0.05 = $ 310 /yr

Electric Motor Drive Performance

Worksheet 14-E1

(Page 1 of 1)

Company: WORKED EXAMPLE #1(SHT.1) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Rated voltage, V_r 2300 volts

Rated full load current, I_r 127 amps

Measured voltage, V 2334 volts

Measured current, I 121 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate motor shaft power output 450 kW

Rated full load power factor, $p.f._r$ 0.92 (decimal)

Measured power factor, p.f. 0.92 (decimal)

Unit electrical energy cost, Ce $ N/A /kWh

Operation time, h N/A h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{121 \times 2334 \times 0.92}{127 \times 2300 \times 0.92}$$

$= 0.97$ USE NAMEPLATE EFFICIENCY

Motor efficiency, Ef_m (~~Figure E-1~~) (NAMEPLATE) 0.95 (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{2334 \times 121 \times 1.73 \times 0.92}{1000}$$

$= 449$ kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m = 449 \times 0.95 = 427$ kW

Annual energy cost saving = $Wm_i \times h \times Ce$ = x x = $ N/A /yr

Compressor Drive

Worksheet 14-C4
(Page 1 of 1)

Company: WORKED EXAMPLE #1 (SHT. 2) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Motor shaft power output, Wm_o (FROM WORKSHEET 14-E1)	427	kW
Turbine shaft power output, Wt_o	N/A	kW
Drive loss (Table 1)	2.3	%

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$$= 1 - \frac{2.3}{100}$$

$$= 0.977$$

Power input to the compressor shaft, $Wc_i = Wm_o \times Ef_d$

~~or, $Wc_i = Wt_o \times Ef_d$~~

$$= 427 \times 0.977$$

$$= 417 \text{ kW}$$

Compressor Component Heating/Cooling
Worksheet 14-C5
(Page 1 of 2)

Company: WORKED EXAMPLE #1 (SHT. 3) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Power input to the compressor shaft, Wc_i 417 kW (WORK SHT. 14-C4)

Cooling media WATER

Type of fuel No. 2 OIL

Fuel higher heating value, HHV (Appendix C) 38.68 MJ/ L

Unit fuel, steam or electrical energy cost, C_f $ 0.40 / L

Operation time, h 4480 h/yr

Efficiency of fuel burning device, Ef_b 0.75 (decimal)
(if unknown, use 0.75)

Portion of input energy released

Component	Heat Exchange Efficiency Table 2
COMPRESSOR	0.04
INTERCOOLER	0.44
Total, Ef_e	0.48

Potential heat recovery, $Q_{pr} = 3.6 \times Wc_i \times Ef_e$

$= 3.6 \times 417 \times 0.48$

$= 721$ MJ/h

Compressor Component Heating/Cooling

Worksheet 14-C5
(Page 2 of 2)

Company: WORKED EXAMPLE #1 (SHT. 4) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Annual fuel ~~cost~~ saving $= \dfrac{Q_{pr} \times Cf \times h}{HHV \times Ef_b}$

$$= \frac{721 \times 0.40 \times 4480}{38.68 \times 0.75}$$

= \$ 44,537 /yr

Annual steam cost saving $= \dfrac{Q_{pr} \times Cs \times h}{2.2}$

$$= \frac{\quad \times \quad \times \quad}{2.2}$$

= \$ N/A /yr

Annual cooling energy cost saving (if Sc is unknown use 80 kWh/GJ for an electrical cooling system and 0.45 kg/GJ for a system operated by steam)

Annual Savings $= \dfrac{Q_{pr} \times C\_ \times Sc \times h}{1000}$

$$= \frac{\quad \times \quad \times \quad \times \quad}{1000}$$

= \$ N/A /yr

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE #2 (SHT. 1) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Rated voltage, V_r	575	volts
Rated full load current, I_r	135	amps
Measured voltage, V	580	volts
Measured current, I	109	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate shaft power output	112	kW
Rated full load power factor, $p.f._r$	0.89	(decimal)
Measured power factor, p.f.	0.90	(decimal)
Unit electrical energy cost, Ce	$ N/A	/kWh
Operation time, h	N/A	h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{109 \times 580 \times 0.90}{135 \times 575 \times 0.89}$$

$$= 0.82$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 109 \times 1.73 \times 0.90}{1000}$$

$$= 98.4 \text{ kW}$$

Motor shaft power output, $Wm_o = Wm_i \times Ef_m = 98.4 \times 0.92 = 90.5$ kW

Annual energy cost saving = Wm_i x h x Ce = x x = $ N/A /yr

Compressor Drive
Worksheet 14-C4
(Page 1 of 1)

Company: WORKED EXAMPLE #2 (SHT. 2) Date: 86/08/19
Location: COMPRESSORS - RETROFIT By: MBE

Data

Motor shaft power output, Wm_o (WORKSHEET 14-E1) 90.5 kW

Turbine shaft power output, Wt_o N/A kW

Drive loss (Table 1) 3.3 %

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$= 1 - \frac{3.3}{100}$

= 0.967

Power input to the compressor shaft, $Wc_i = Wm_o \times Ef_d$

~~or, $Wc_i = Wt_o \times Ef_d$~~

= 90.5 x 0.967

= 87.5 kW

Compressor Component Heating/Cooling

Worksheet 14-C5

(Page 1 of 2)

Company: WORKED EXAMPLE #2 (SHT. 3) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Power input to the compressor shaft, Wc_i (WORKSHEET 14-C4) 87.5 kW

Cooling media AIR

Type of fuel ELECTRICITY

Fuel higher heating value, HHV (Appendix C) N/A MJ/ ____

Unit fuel, steam or electrical energy cost, C e $ 0.05 / kWh

Operation time, h 1200 h/yr

Efficiency of fuel burning device, Ef_b (if unknown, use 0.75) N/A (decimal)

Portion of input energy released

Component	Heat Exchange Efficiency Table 2
COMPRESSOR	0.05
INTER COOLER	0.45
V-BELT DRIVE	0.03
MOTOR	0.08
Total, Ef_e	0.61

Potential heat recovery, $Q_{pr} = 3.6 \times Wc_i \times Ef_e$

$= 3.6 \times 87.5 \times 0.61$

$=$ 192 MJ/h

Compressor Component Heating/Cooling

Worksheet 14-C5
(Page 2 of 2)

Company: WORKED EXAMPLE #2 (SHT. 4) Date: 86 / 08 / 19

Location: COMPRESSORS - RETROFIT By: MDE

Annual fuel cost saving $= \dfrac{Q_{pr} \times Cf \times h}{HHV \times Ef_b}$

$= \dfrac{\quad \times \quad \times \quad}{\quad \times \quad}$

= $ N/A /yr

Annual steam cost saving $= \dfrac{Q_{pr} \times Cs \times h}{2.2}$

$= \dfrac{\quad \times \quad \times \quad}{2.2}$

= $ N/A /yr

Annual cooling energy ~~cost~~ saving (if Sc is unknown use 80 kWh/GJ for an electrical cooling system and 0.45 kg/GJ for a system operated by steam)

Annual Savings $= \dfrac{Q_{pr} \times C_e \times Sc \times h}{1000}$

$= \dfrac{192 \times 0.05 \times 80 \times 1200}{1000}$

= $ 922 /yr

Compressor Component Heating/Cooling

Worksheet 14-C5
(Page 1 of 2)

Company: WORKED EXAMPLE #3 (SHT. 1) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Power input to the compressor shaft, Wc_i (4 × 75) 300 kW

Cooling media { AIR FOR COMPRESSORS & INTERCOOLERS
WATER FOR AFTERCOOLER

Type of fuel Nº 6 OIL

Fuel higher heating value, HHV (Appendix C) 40.5 MJ/ L

Unit fuel, steam or electrical energy cost, C_f $ 0.22 / L

Operation time, h 2160 h/yr

Efficiency of fuel burning device, Ef_b
(if unknown, use 0.75) 0.75 (decimal)

Portion of input energy released

Component	Heat Exchange Efficiency Table 2
MOTOR	0.08
INTERCOOLERS	0.45
COMPRESSORS	0.05
Total, Ef_e	0.58

Potential heat recovery, $Q_{pr} = 3.6 \times Wc_i \times Ef_e$

$= 3.6 \times 300 \times 0.58$

$=$ 626 MJ/h

Compressor Component Heating/Cooling

Worksheet 14-C5
(Page 2 of 2)

Company: WORKED EXAMPLE #3 (SHT. 2) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Annual fuel ~~cost~~ saving $= \dfrac{Q_{pr} \times Cf \times h}{HHV \times Ef_b}$

$= \dfrac{626 \times 0.22 \times 2160}{40.5 \times 0.75}$

= \$ 9,793 /yr

Annual steam cost saving $= \dfrac{Q_{pr} \times Cs \times h}{2.2}$

$= \dfrac{\quad \times \quad \times \quad}{2.2}$

= \$ N/A /yr

Annual cooling energy cost saving (if Sc is unknown use 80 kWh/GJ for an electrical cooling system and 0.45 kg/GJ for a system operated by steam)

Annual Savings $= \dfrac{Q_{pr} \times C\_ \times Sc \times h}{1000}$

$= \dfrac{\quad \times \quad \times \quad \times \quad}{1000}$

= \$ N/A /yr

Electric Motor Drive Performance

Worksheet 14-E1

(Page 1 of 1)

Company: WORKED EXAMPLE #4 (SHT. 1) Date: 86/08/19

Location: COMPRESSORS - RETROFIT (LOADED) By: MBE

Rated voltage, V_r	N/A	volts
Rated full load current, I_r	N/A	amps
Measured voltage, V (V_L)	584	volts
Measured current, I (I_L)	419	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	N/A	(decimal)
Measured power factor, p.f. $(p.f._L)$	0.86	(decimal)
Unit electrical energy cost, Ce	$ 0.05	/kWh
Operation time, h (h_L)	6596	h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) ______ (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{584 \times 419 \times 1.73 \times 0.86}{1000}$$

= 364 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = N/A kW

Annual energy cost ~~saving~~ = $Wm_i \times h \times Ce$ = 364 x 6596 x 0.05 = $ 120,047 /yr (WHILE LOADED)

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE #4 (SHT. 2) Date: 86/08/19
Location: COMPRESSORS - RETROFIT UNLOADED By: MBE

Rated voltage, V_r N/A volts

Rated full load current, I_r N/A amps

Measured voltage, V (V_u) 587 volts

Measured current, I (I_u) 251 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate shaft power output N/A kW

Rated full load power factor, $p.f._r$ N/A (decimal)

Measured power factor, p.f. $(p.f._u)$ 0.32 (decimal)

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h (h_u) 1828 h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{587 \times 251 \times 1.73 \times 0.32}{1000}$$

= 82 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = N/A kW

Annual energy cost ~~saving~~ = $Wm_i \times h \times Ce$ = 82 x 1828 x 0.05 = $ 7,495 /yr
(WHILE UNLOADED)

Equivalent Free Air Flow

Worksheet 14-C6

Page 1 of 1

Company: WORKED EXAMPLE #4 (SHT. 3) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Initial air flow rate, f_{a1}	1385	L/s
Initial temperature, T_1	317.15	K
Initial pressure, P_1	156.325	kPa(absolute)

$$\text{Equivalent free air flow rate, } f_{as} = \frac{P_1 \times f_{a1} \times 293.15}{T_1 \times 101.325}$$

$$= \frac{156.325 \times 1385 \times 293.15}{317.15 \times 101.325}$$

$$= 1975 \text{ L/s}$$

Compressor Drive

Worksheet 14-C4
(Page 1 of 1)

Company: WORKED EXAMPLE #4 (SHT. 4) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Motor shaft power output, Wm_o To Be Calculated kW

Turbine shaft power output, Wt_o N/A kW

Drive loss (Table 1) (APPROXIMATE BASED ON Wc_i) 3.1 %

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$= 1 - \frac{3.1}{100}$

$=$ 0.969

Power input to the compressor shaft, $Wc_i = Wm_o \times Ef_d$

~~or, $Wc_i = Wt_o \times Ef_d$~~

$132 = Wm_o \times 0.969$

$Wm_o =$ 136 kW

Motor Replacement

Worksheet 14-E2
(Page 1 of 1)

Company: WORKED EXAMPLE #4 (SHT. 5) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Motor Data:

Initial motor power input, Wm_{i1} N/A kW

Required motor shaft power output, Wm_o (WORKSHEET 14-C4) 136 kW

Replacement motor rated shaft power output 149 kW (1)

Operation time, h 8424 h/yr

Unit electrical energy cost, Ce $ 0.05 /kWh

$$\text{Load ratio} = \frac{Wm_o}{(1)}$$

$$= \frac{136}{149}$$

$$= 0.91$$

Replacement motor efficiency, Ef_m 0.92 (decimal)
(~~Nameplate~~ or Figure E-1)

$$\text{Replacement motor power input, } Wm_{i2} = \frac{Wm_o}{Ef_m}$$

$$= \frac{136}{0.92}$$

$$= 148 \text{ kW}$$

Annual energy cost ~~savings~~ = $(Wm_{i1} - Wm_{i2}) \times h \times Ce$

= (0 − 148) x 8424 x $0.05

= $(−) 62,338 /yr
(EXTRA COST)

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE #5 (SHT.1) Date: 86/08/19

Location: COMPRESSORS-RETROFIT By: MBE

Rated voltage, V_r	N/A	volts
Rated full load current, I_r	N/A	amps
Measured voltage, V	233	volts
Measured current, I	58	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	N/A	(decimal)
Measured power factor, p.f.	0.86	(decimal)
Unit electrical energy cost, Ce	$ N/A	/kWh
Operation time, h	N/A	h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\_\_\_ \times \_\_\_ \times \_\_\_}{\_\_\_ \times \_\_\_ \times \_\_\_}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{233 \times 58 \times 1.73 \times 0.86}{1000}$$

= 20.1 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = N/A kW

Annual energy cost saving = $Wm_i \times h \times Ce$ = x x = $ N/A /yr

Equivalent Free Air Flow

Worksheet 14-C6
Page 1 of 1

Company: WORKED EXAMPLE #5 (SAT.2) Date: 86/08/19

Location: COMPRESSORS-RETROFIT By: MBE

Data

Initial air flow rate, f_{a1} 50 L/s

Initial temperature, T_1 289.15 K

Initial pressure, P_1 101.325 kPa(absolute)

$$\text{Equivalent free air flow rate, } f_{as} = \frac{P_1 \times f_{a1} \times 293.15}{T_1 \times 101.325}$$

$$= \frac{101.325 \times 50 \times 293.15}{289.15 \times 101.325}$$

$$= 51 \text{ L/s}$$

Motor Replacement

Worksheet 14-E2
(Page 1 of 1)

Company: WORKED EXAMPLE #5 (SHT. 3) Date: 86/08/19
Location: COMPRESSORS - RETROFIT By: MBE

Motor Data

Initial motor power input, Wm_{i1} (WORKSHEET 14-E1) 20.1 kW
Required motor shaft power output, Wm_o 14.7 kW
Replacement motor rated shaft power output 18.6 kW (1)
Operation time, h 4000 h/yr
Unit electrical energy cost, Ce $ 0.05 /kWh

$$\text{Load ratio} = \frac{Wm_o}{(1)}$$

$$= \frac{14.7}{18.6}$$

$$= 0.79$$

Replacement motor efficiency, Ef_m 0.92 (decimal)
(~~Nameplate~~ or Figure E-1)

$$\text{Replacement motor power input, } Wm_{i2} = \frac{Wm_o}{Ef_m}$$

$$= \frac{14.7}{0.92}$$

$$= 16.0 \text{ kW}$$

Annual energy ~~cost~~ savings = $(Wm_{i1} - Wm_{i2}) \times h \times Ce$

= (20.1 − 16.0) x 4000 x $0.05

= $ 820 /yr

Compressor Drive
Worksheet 14-C4
(Page 1 of 1)

Company: WORKED EXAMPLE #6 (SIT. 1) Date: 86/08/19
Location: COMPRESSORS - RETROFIT By: MBE

Data

Motor shaft power output, Wm_o	75	kW
Turbine shaft power output, Wt_o	N/A	kW
Drive loss (Table 1)	3.3	%

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$$= 1 - \frac{3.3}{100}$$

$$= 0.967$$

Power input to the compressor shaft, $Wc_i = Wm_o \times Ef_d$

~~or, $Wc_i = Wt_o \times Ef_d$~~

$$= 75 \times 0.967$$

$$= 72.5 \text{ kW}$$

Compressor Component Heating/Cooling

Worksheet 14-C5
(Page 1 of 2)

Company: WORKED EXAMPLE #6 (SHT. 2) Date: 86/08/19
Location: COMPRESSORS - RETROFIT By: MBE

Data

Power input to the compressor shaft, Wc_i (WORKSHEET 14-C4) 72.5 kW

Cooling media AIR

Type of fuel STEAM

Fuel higher heating value, HHV (Appendix C) N/A MJ/ ____

Unit fuel, steam or electrical energy cost, C_s $ 0.22 / kg

Operation time, h 2000 h/yr

Efficiency of fuel burning device, Ef_b (if unknown, use 0.75) N/A (decimal)

Portion of input energy released

Component	Heat Exchange Efficiency Table 2
AFTERCOOLER	0.45
Total, Ef_e	0.45

Potential heat recovery, $Q_{pr} = 3.6 \times Wc_i \times Ef_e$

$= 3.6 \times 72.5 \times 0.45$

$=$ 117 MJ/h

Compressor Component Heating/Cooling

Worksheet I4-C5

(Page 2 of 2)

Company: WORKED EXAMPLE #6 (SHT 3) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Annual fuel cost saving $= \dfrac{Q_{pr} \times Cf \times h}{HHV \times Ef_b}$

$= \dfrac{\_\_\_ \times \_\_\_ \times \_\_\_}{\_\_\_ \times \_\_\_}$

= $ N/A /yr

Annual steam ~~cost~~ saving $= \dfrac{Q_{pr} \times Cs \times h}{2.2}$

$= \dfrac{117 \times 0.022 \times 2000}{2.2}$

= $ 2,340 /yr

Annual cooling energy cost saving (if Sc is unknown use 80 kWh/GJ for an electrical cooling system and 0.45 kg/GJ for a system operated by steam)

Annual Savings $= \dfrac{Q_{pr} \times C\_ \times Sc \times h}{1000}$

$= \dfrac{\_\_\_ \times \_\_\_ \times \_\_\_ \times \_\_\_}{1000}$

= $ N/A /yr

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE #7 (SHT. 1) Date: 86/08/19

Location: COMPRESSORS - RETROFIT (LOADED) By: MBE

Rated voltage, V_r 575 volts

Rated full load current, I_r 91.2 amps

Measured voltage, V (V_L) 575 volts

Measured current, I (I_L) 90 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate shaft power output 75 kW

Rated full load power factor, $p.f._r$ 0.91 (decimal)

Measured power factor, p.f. ($p.f._L$) 0.90 (decimal)

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h (h_L) 3072 h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{90 \times 575 \times 0.90}{91.2 \times 575 \times 0.91}$$

$$= 0.98$$

Motor efficiency, Ef_m (Figure E-1) 0.92 (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 90 \times 1.73 \times 0.90}{1000}$$

$$= 80.6 \text{ kW}$$

(= Wm_{oL} FOR WORKSHEET 14-C7)

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = 80.6 x 0.92 = 74.2 kW

Annual energy cost ~~saving~~ = Wm_i x h x Ce = 80.6 x 3072 x 0.05 = $ 12,380 /yr
(WHILE LOADED)

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: WORKED EXAMPLE #7 (SHT. 2) Date: 86/08/19

Location: COMPRESSORS - RETROFIT (UNLOADED) By: MBE

Rated voltage, V_r N/A volts

Rated full load current, I_r N/A amps

Measured voltage, V (V_u) 580 volts

Measured current, I (I_u) 32.7 amps

Phase function, Y 1.73
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate shaft power output N/A kW

Rated full load power factor, $p.f._r$ N/A (decimal)

Measured power factor, p.f. ($p.f._u$) 0.71 (decimal)

Unit electrical energy cost, Ce $ 0.05 /kWh

Operation time, h (h_u) 3881 h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{580 \times 32.7 \times 1.73 \times 0.71}{1000}$$

= 23.3 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = N/A kW

Annual energy cost ~~saving~~ = Wm_i x h x Ce = 23.3 x3881 x0.05 = $ 4,521 /yr
(WHILE UNLOADED)

Variable Speed Drive

Worksheet 14-C7

(Page 1 of 2)

Company: WORKED EXAMPLE #7 (SM7.3) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Data

Motor shaft power output-loaded, Wm_{oL} (FROM WORKSHEET 14-E1 LOADED) 74.2 kW

Rated motor efficiency, Ef_{mr} 0.92 (decimal)

Equivalent operation time — fully loaded, h_L 3072 h/yr

Operation time at variable speed, h_v (3072 + 3881) 6953 h/yr

Efficiency of variable speed controller, Ef_{vc} 0.95 (decimal)

Compressor drive loss (Table 1) 3.6 %

Unit electrical energy cost, Ce $ 0.05 /kWh

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$$= 1 - \frac{3.6}{100}$$

$$= 0.964$$

Power input to compressor shaft – fully loaded,

$$Wc_{iL} = Wm_{oL} \times Ef_d$$

$$= 74.2 \times 0.964$$

$$= 71.5 \text{ kW}$$

Average power input to compressor shaft, $Wc_{iv} = \frac{Wc_{iL} \times h_L}{h_v}$

$$= \frac{71.5 \times 3072}{6953}$$

$$= 31.6 \text{ kW}$$

Variable Speed Drive
Worksheet 14-C7
(Page 2 of 2)

Company: WORKED EXAMPLE #7 (SAT. 4) Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Average motor power input, $Wm_{iv} = \dfrac{Wc_{iv}}{Ef_m \times Ef_{vd} \times Ef_d}$

$$= \frac{31.6}{0.92 \times 0.95 \times 0.964}$$

$$= 37.5 \text{ kW}$$

Annual energy cost $= Wm_{iv} \times h_v \times Ce$

$$= 37.5 \times 6953 \times 0.05$$

$$= \$13,037/\text{yr}$$

Electric Motor Drive Performance

Worksheet 14-E1

(Page 1 of 1)

Company: WORKED EXAMPLE #10 Date: 86/08/19

Location: COMPRESSORS - RETROFIT By: MBE

Rated voltage, V_r	N/A	volts
Rated full load current, I_r	N/A	amps
Measured voltage, V	230	volts
Measured current, I	21	amps
Phase function, Y (1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)	1.73	
Nameplate shaft power output	N/A	kW
Rated full load power factor, $p.f._r$	N/A	(decimal)
Measured power factor, p.f.	0.79	(decimal)
Unit electrical energy cost, Ce	$ 0.05	/kWh
Operation time, h	1102	h/yr

Load ratio $= \dfrac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \dfrac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$

= N/A

Motor efficiency, Ef_m (Figure E-1) N/A (decimal)

Motor power input, $Wm_i = \dfrac{V \times I \times Y \times p.f.}{1000}$

$= \dfrac{230 \times 21 \times 1.73 \times 0.79}{1000}$

= 6.6 kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m =$ x = N/A kW

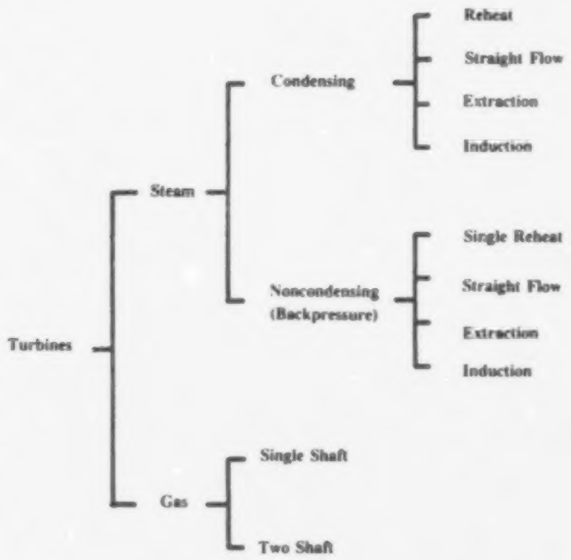
Annual energy cost saving = $Wm_i \times h \times Ce$ = 6.6 x1102 x0.05 = $ 364 /yr

SECTION 2 — TURBINES

FUNDAMENTALS

SAVE
ENERGY

Turbines are rotary machines that convert expanding hot gas or vapor into shaft power. Figure 19 is a classification of the various types of turbines that are operated in Industrial, Commercial and Institutional facilities. The two major classifications are *steam* and *gas* turbines. Because of their differing energy sources and operating characteristics, steam and gas turbines will be addressed separately. For the purposes of this text, steam turbines will be discussed first.


Turbines
Steam
Gas
Condensing
Noncondensing (Backpressure)
Reheat
Straight Flow
Extraction
Induction
Single Reheat
Straight Flow
Extraction
Induction
Single Shaft
Two Shaft

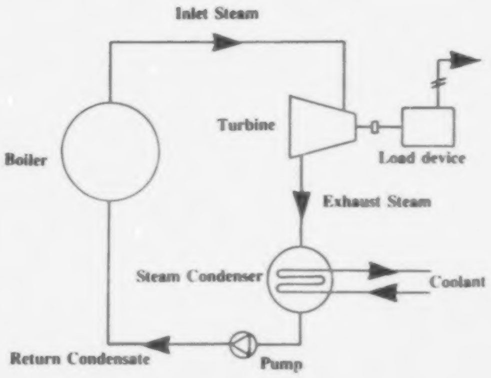
Classification of Turbines
Figure 19

Steam Turbine Types

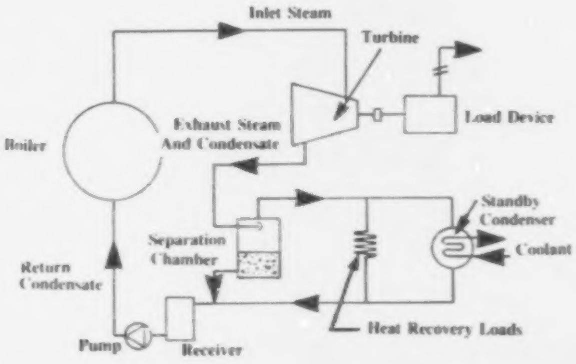
Steam turbines are classified by the condition of the exhaust steam as condensing and noncondensing.

Condensing turbines (Figure 20) usually utilize superheated inlet steam to minimize condensation within the turbine, and operate with the exhaust steam below atmospheric pressure. The low exhaust steam pressure is created by an external heat exchanger that uses exterior cooling to condense the steam as it leaves the turbine.

Noncondensing turbines (Figure 21), sometimes called *backpressure turbines*, operate with the exhaust steam at or above atmospheric pressure. They often utilize inlet steam at saturation pressure and temperature resulting in a mixture of steam and condensate, or *wet*, exhaust steam.



Condensing Steam Turbine System
Figure 20



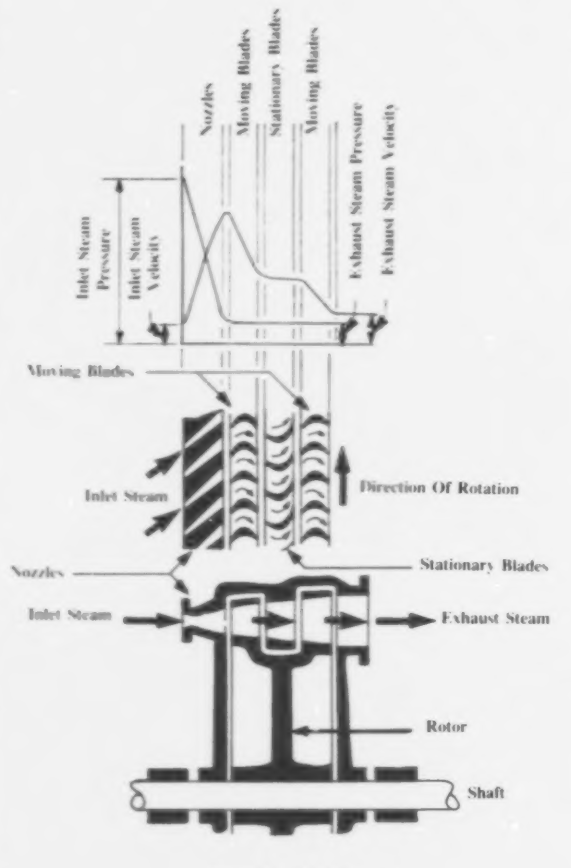
Noncondensing Steam Turbine System
Figure 21

Steam Turbine Operation

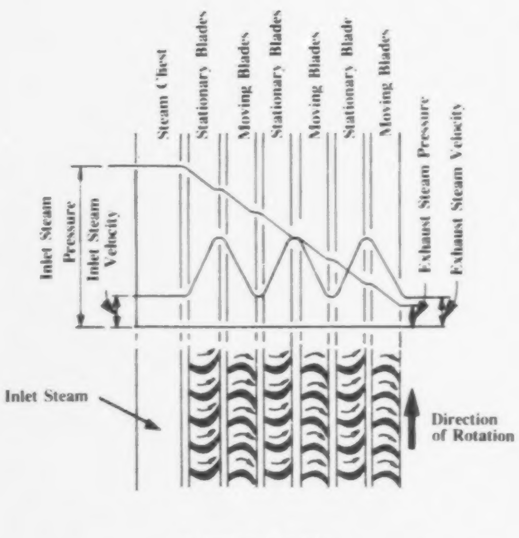
A steam turbine converts the energy in steam to shaft power by impingement against, or flow over, a series of blades attached to the shaft. In a pure impulse turbine (Figure 22), steam passes through a series of nozzles which expand the steam causing a reduction in the pressure and an increase in the velocity. The pressure and velocity profiles in Figure 22 depict the changes within the turbine. High velocity steam is directed into the first row of moving blades. The blades absorb part of the velocity and rotate the shaft. The steam then passes through a row of stationary blades that redirect the steam to the second row of moving blades where the steam velocity is further reduced to extract more energy. The cumulative effect of the reaction forces on all of the moving blades provides the rotational power to the shaft. A multistage turbine may contain several rows of stationary and moving blades. A single stage unit would have no stationary blades.

In a pure reaction turbine (Figure 23), steam passes through an initial stage of stationary blades which reduces the pressure and accelerates the steam against the first stage of moving blades. The process is repeated through subsequent stages of stationary and moving blades to provide the rotational power to the shaft.

In actual applications most turbine blades function under a combination of impulse and reaction effect. Those blades which are short and use high pressure steam are mostly impulse driven, and those which are longer and use low pressure steam are mostly reaction driven. The majority of steam turbines used to drive compressors, fans, pumps and small generation machines have one to three sets of mostly impulse blades.


Inlet Steam Pressure
Inlet Steam Velocity
Nozzles
Moving Blades
Stationary Blades
Moving Blades
Exhaust Steam Pressure
Exhaust Steam Velocity
Moving Blades
Inlet Steam
Direction Of Rotation
Nozzles
Stationary Blades
Inlet Steam
Exhaust Steam
Rotor
Shaft

Impulse Turbine
Figure 22


Inlet Steam Pressure
Inlet Steam Velocity
Steam Chest
Stationary Blades
Moving Blades
Stationary Blades
Moving Blades
Stationary Blade
Moving Blades
Exhaust Steam Pressure
Exhaust Steam Velocity
Inlet Steam
Direction of Rotation

Reaction Turbine
Figure 23

Thermal Properties of Steam

The fundamentals of steam and condensate are discussed in Steam and Condensate Systems, Module 8. Those properties which relate directly to steam turbines will be discussed here.

Specific Enthalpy is the amount of heat that a substance contains per unit of mass. It is designated by the notation *h* and has units of kilojoules per kilogram (kJ/kg) of steam.

Entropy is the rate of change of enthalpy per unit of temperature change. It is designated by the notation *s* and has units of kilojoules per kilogram of steam per degree Celsius [kJ/(kg·°C)].

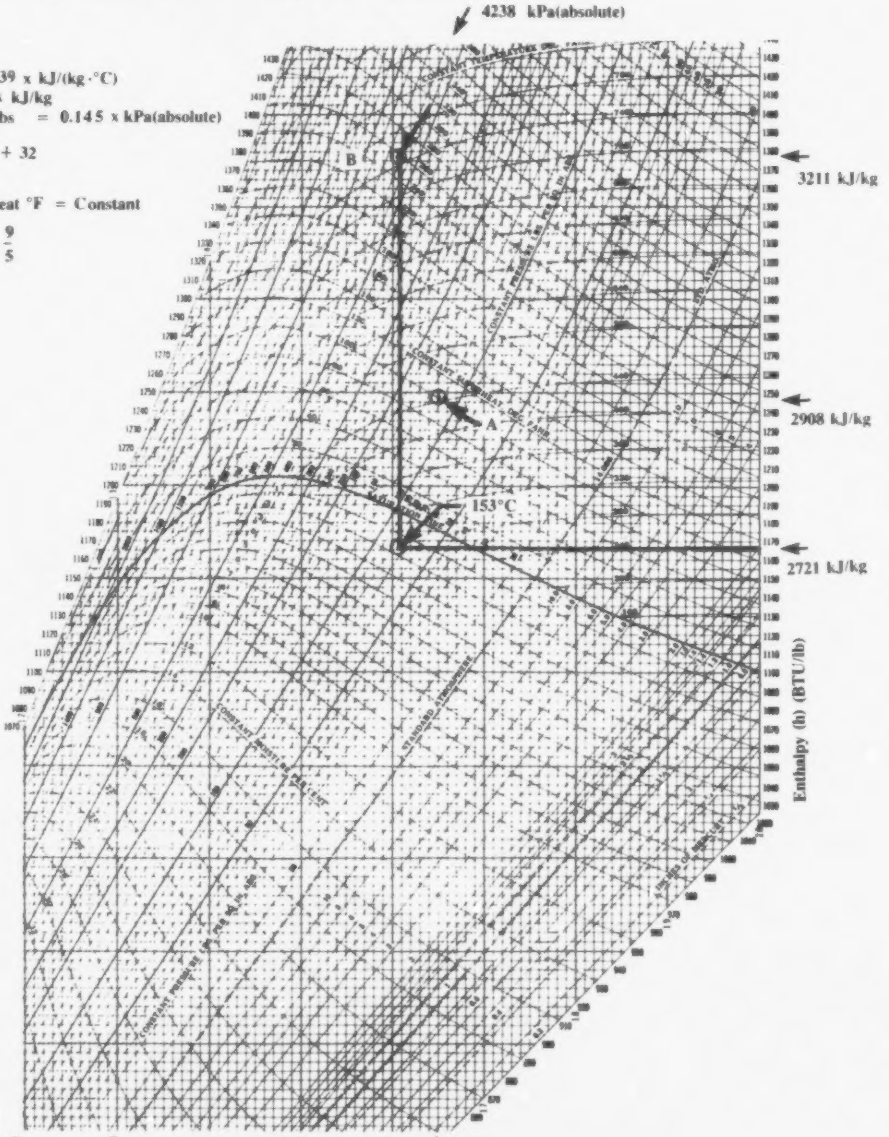
Steam tables (Table 6) are tabulated listings of the following thermal properties of water and steam at various sustained temperatures and pressures.

- The *specific volume*, or reciprocal of density, for water at saturation temperature (v_f) and saturated steam (v_g).
- The enthalpy for water at saturation temperature (h_f), evaporation or change of state (h_{fg}), and saturated steam (h_g).
- The entropy for water at saturation temperature (s_f), evaporation or change of state (s_{fg}), and saturated steam (s_g).

The values for saturated steam are for dry steam as pure vapor without entrained water droplets. The theoretical values can be used for most steam system calculations. Although most systems operate close to these theoretical values, some water is usually present. The amount of water present is a measure of the steam "quality" and is expressed as *constant moisture per cent* by weight.

The *Mollier Diagram* (Figure 24) is a graphical representation of the steam tables plotted with enthalpy (h) on the vertical scale and entropy (s) on the horizontal scale. Also included on the chart are absolute pressures, temperatures, constant moisture per cent and constant superheat temperatures. The major portions of the diagram are separated by the *saturation line*; below which, the steam contains some water droplets and, above which, the steam is superheated and acts as a gas. Superheated steam is obtained by adding more heat to pure (saturated) steam resulting in a temperature rise. If superheating at constant pressure is continued the volume of the steam increases. Below the saturation line the temperature remains constant for a constant pressure.

The Mollier diagram is used to determine the characteristics of steam, usually from temperature and pressure information. For example, to determine the properties of steam at 862 kPa(absolute) and 55°C superheat, perform the necessary conversions, locate the 125 "LBS. PER SQ. IN. ABS." line and move to the intersection with the 99 "CONSTANT SUPERHEAT, DEG. FAHR." line; identified as point A. Moving horizontally, the enthalpy (h) is 2908 kJ/kg. Moving vertically, the entropy (s) is 6.91 kJ/(kg·°C).



Conversions:

BTU/(lb·°F) = 0.239 x kJ/(kg·°C)

BTU/lb = 0.429 x kJ/kg

Lbs Per Sq. In. Abs = 0.145 x kPa(absolute)

$$°F = \left(°C \times \frac{9}{5}\right) + 32$$

$$\text{Constant Superheat } °F = \text{Constant Superheat } °C \times \frac{9}{5}$$

Partial Mollier Diagram

Figure 24

Steam Turbine Performance Measurement

To analyze turbine performance, it is necessary to measure the inlet and exhaust steam conditions, determine the design characteristics of the turbine and the driven devices, and measure the steam flow rate. If a steam flow meter is not available, the steam flow rate can be estimated by measuring the quantity of condensate produced at the end of the total system during a measured period of time. The condensate flow rate from the exhaust steam separator can be measured by the same method.

Steam Turbine Performance

Manufacturers' performance information is the most reliable source of data for steam turbine operating characteristics. Because of the number of variables affecting performance, the data normally must be obtained from the manufacturer for each particular application. Published catalogue tables usually list only maximum power outputs and maximum speeds for certain inlet and outlet conditions. When manufacturers' data is not available, the energy consumption of a particular steam turbine can be estimated using measured data and the equations that are discussed in this section.

The conversion of the energy in steam to mechanical work in a turbine is expressed by the terms, *ideal steam rate*, *actual steam rate*, and *mechanical steam turbine efficiency*.

Ideal Steam Rate

The ideal steam rate is the quantity of steam that will produce a unit of shaft energy output from an ideal steam turbine having no machinery or thermal losses, and is represented by the following equation.

$$\text{I.S.R.} = \frac{3600}{h_i - h_o}$$

Where, I.S.R. = ideal steam rate (kg/kWh)

h_i = enthalpy of inlet steam (kJ/kg)

h_o = enthalpy of outlet steam (at entropy of the inlet steam) (kJ/kg)

3600 = conversion of kWh to kJ

An ideal steam turbine converts energy in the steam to mechanical shaft energy at a uniform rate, or constant entropy. In reference to the Mollier diagram, this would be represented by a drop in enthalpy along a constant entropy line. Such a line is shown on Figure 24 (Points B to C) for an analysis of a steam turbine. However, heat loss in an actual turbine causes some change in entropy during the energy exchange.

The ratio of ideal steam rates can be used to estimate the effect of a change in inlet or outlet conditions for a particular turbine.

$$\text{Thus, } (\text{I.S.R.})_2 = (\text{I.S.R.})_1 \times \frac{h_{i1} - h_{o1}}{h_{i2} - h_{o2}}$$

Where, $(\text{I.S.R.})_1$, $(\text{I.S.R.})_2$ = initial and revised ideal steam rates (kg/kWh)

h_{i1}, h_{i2} = initial and revised enthalpies of inlet steam (kJ/kg)

h_{o1}, h_{o2} = initial and revised enthalpies of outlet steam (at entropies of inlet steam) (kJ/kg)

Actual Steam Rate

The actual steam rate, usually referred to as *steam rate*, is the flow of steam required to produce a particular shaft power output. Steam rates are valuable for estimating the effect of changes to a particular turbine, as shown by Table 7. In the table the steam rate is expressed in kilograms per kilowatt hour and per cent of the manufacturers' design steam rates for a particular set of inlet and exhaust conditions. For example, the first section of the table shows the steam rate for an inlet condition of 850 kPa(gauge) and 110°C superheat to be 88 per cent of the steam rate for 850 kPa and 0°C superheat.

Manufacturers' design steam rates for steam turbines equipped with speed reducers or increasers include the 2 to 3 per cent loss in shaft power associated with the drive. The actual output power rating of a steam turbine represents the power at the gear box output shaft. Actual steam rate is expressed by the following equation.

$$\text{A.S.R.} = \frac{f_s}{Wt_o}$$

Where, A.S.R. = actual steam rate (kg/kWh)

f_s = steam flow rate (kg/h)

Wt_o = turbine shaft power output (kW)

The ratio of ideal steam rates can also be used to estimate the effect of a change in inlet or outlet conditions on actual steam rate.

$$\text{Thus, } (\text{A.S.R.})_2 = (\text{A.S.R.})_1 \times \frac{h_{i1} - h_{o1}}{h_{i2} - h_{o2}}$$

Where, $(\text{A.S.R.})_1$, $(\text{A.S.R.})_2$ = initial and revised actual steam rates (kg/kWh)

For a particular shaft power output the equation can be restated to calculate the effect on steam flow rate.

$$f_{s2} = f_{s1} \times \frac{h_{i1} - h_{o1}}{h_{i2} - h_{o2}}$$

Where, f_{s1}, f_{s2} = initial and revised steam flow rates (kg/h)

The annual steam cost saving ($/yr) can be calculated by the following equation.

Annual steam cost saving = $(f_{s1} - f_{s2}) \times h \times Cs$

where, h = operation time (h/yr)

Cs = unit steam cost ($/kg)

Although the calculation is valid for all steam turbines, *the value cannot be used for comparison of different types of steam turbines* because it is related to steam flow rate only and does not account for steam conditions at the inlet.

Mechanical Steam Turbine Efficiency

The mechanical steam turbine efficiency is the ratio of the ideal steam rate to the actual steam rate and is expressed by the following equation.

$$Ef_t = \frac{\text{I.S.R.}}{\text{A.S.R.}} \times 100$$

Where, Ef_t = mechanical steam turbine efficiency (%)

Typical mechanical efficiencies, based on the actual steam rate, range from 55 to 80 per cent. The mechanical steam turbine efficiency increases with a greater enthalpy difference between steam at the inlet and at the outlet, and with greater number of stages, larger machines and higher speeds. The mechanical steam turbine efficiency includes any losses associated with a final drive gear box. The selection of a turbine for a particular application should always take into account the potential for utilizing the energy remaining in the exhaust steam.

Steam Turbine System Peformance

The total energy input required for a steam turbine must be defined relative to the total steam and condensate system in which the turbine operates. Many condensing turbine systems (Figure 20) reject their condenser heat to a cooling tower or large body of water in order to maintain the lowest possible exhaust steam temperature and maximum turbine power output. In such a system the energy input required for the turbine equals the total difference in enthalpy between the inlet steam and the condensate returned to the boiler.

In a non-condensing (back pressure) turbine system (Figure 21) the exhaust steam and condensate mixture is usually passed through a separation chamber (flash tank) where the steam is drawn off for other uses. In such a system the energy input required for the turbine equals the enthalpy of the inlet steam minus the enthalpy of the exhaust steam and condensate mixture. This system makes more efficient use of the total steam energy than the condensing turbine system provided that all of the separated steam is used and the condensate is returned to the boiler. Some condensing turbine installations are also able to utilize the low temperature heat of the condenser coolant for such applications as preheating of air or fluids, or as a heat source for heat pumps.

An approximation of the energy that is extracted from the steam to produce shaft energy is provided by the following equation.

$$Q_e = \frac{f_s \times (h_i - h_e)}{1\,000\,000}$$

Where, Q_e = energy extracted from the steam (GJ/h)

f_s = steam flow rate (kg/h)

h_i = enthalpy of inlet steam (kJ/kg)

h_e = enthalpy of outlet steam and condensate at exhaust conditions (kJ/kg)

1 000 000 = conversion from kJ to GJ

The total energy input required for a steam system containing a turbine can be approximated by the following equation.

$$Q_t = \frac{f_s \times (h_i - h_r)}{1\,000\,000}$$

Where, Q_t = total steam system energy input (GJ/h)

h_r = enthalpy of return condensate (kJ/kg)

In many turbine applications the exhaust steam contains a significant amount of condensate. For measurement of turbine performance in a system with heat recovery, it is necessary to separately identify the energy content of the steam and the condensate in the equation for energy extraction.

$$\text{Thus, } Q_e = \frac{(f_s \times h_i) - (f_{co} \times h_{fo}) - (f_{so} \times h_{go})}{1\,000\,000}$$

Where, f_{co} = exhaust condensate flow rate (kg/h)

f_{so} = exhaust steam flow rate (kg/h)

h_{fo} = enthalpy of outlet condensate (at the saturation temperature corresponding to the exhaust steam pressure) (h_f from Table 6) (kJ/kg)

h_{go} = enthalpy of outlet steam (at the exhaust steam saturation pressure) (h_g from Table 6) (kJ/kg)

Since $f_{so} = (f_s - f_{co})$, the equation can be restated to eliminate the need to measure the exhaust steam flow rate, f_{so}.

$$Q_e = \frac{(f_s \times h_i) - (f_{co} \times h_{fo}) - [(f_s - f_{co}) \times h_{go}]}{1\ 000\ 000}$$

The portion of the total steam supply that is attributable to operation of the turbine can be calculated by the following equation.

$$f_{st} = f_s \times \frac{(Q_t - Q_r)}{Q_t}$$

Where, f_{st} = steam flow rate attributable to the turbine (kg/h)

Q_r = energy recovered (for other uses) (GJ/h)

When all of the exhaust steam is condensed for other uses, the equation can be simplified.

$$f_{st} = f_s \times \frac{Q_e}{Q_t}$$

The annual cost ($/yr) of steam attributable to the turbine can be calculated by the following equation.

Annual steam cost = $f_{st} \times h \times Cs$

where, h = operation time (h/yr)

Cs = unit steam cost ($/kg)

Steam Turbine Performance Analysis

An example performance analysis will be carried out on a noncondensing turbine used to drive an alternator and reduce high pressure superheated steam to lower pressure saturated steam. The lower pressure steam is used for heating water and is condensed at 100°C. The turbine is directly connected to the alternator by a shaft coupling.

For this analysis the following data was measured and collected from equipment nameplates.

Inlet steam pressure	4238 kPa(gauge)
Inlet steam temperature	400°C
Outlet steam pressure	515 kPa(gauge)
Outlet steam temperature	153°C
Steam flow rate (measured), f_s	3313 kg/h
Exhaust condensate flow rate (measured), f_{co}	99 kg/h
Unit steam cost, Cs	$0.022/kg
Turbine nameplate shaft power output, Wt_{on}	340 kW
Turbine nameplate steam rate,	10.0 kg/kWh

Measured current (alternator output), I	388 amps
Measured voltage (alternator output), V	575 volts, 3 phase
Alternator efficiency (nameplate), Ef_a	0.92
Measured power factor(alternator output), p.f.	0.80
Operation time, h	8400 h/yr
Unit electrical energy cost, Ce	$0.05/kWh

Appendix E presents the determination of power output (Wa_o) by an electric alternator in terms of the measured output conditions.

$$Wa_o = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{575 \times 388 \times 1.73 \times 0.80}{1000}$$

$$= 309 \text{ kW}$$

The shaft power input to the alternator can also be calculated by the equation from Appendix E.

$$Wa_i = \frac{Wa_o}{Ef_a}$$

$$= \frac{309}{0.92}$$

$$= 336 \text{ kW}$$

Since the turbine is directly connected to the alternator, the drive loss is negligible and the alternator shaft power input (Wa_i) equals the required turbine shaft power output (Wt_o).

As previously defined, the ideal steam rate, I.S.R. $= \dfrac{3600}{h_i - h_o}$

Refer to Figure 24, and read horizontally from the actual turbine inlet conditions.

Enthalpy of inlet steam, $h_i = 3211$ kJ/kg

Move vertically on Figure 24 from the actual turbine inlet conditions to the outlet temperature line, and read horizontally.

Enthalpy of outlet steam, $h_o = 2721$ kJ/kg

$$\text{Therefore, I.S.R.} = \frac{3600}{3211 - 2721}$$

$$= 7.35 \text{ kg/kWh}$$

$$\text{Actual steam rate, A.S.R.} = \frac{f_s}{Wt_o}$$

$$= \frac{3313}{336}$$

$$= 9.86 \text{ kg/kWh}$$

This compares closely with the nameplate steam rate of 10 kg/kWh.

$$\text{The mechanical steam turbine efficiency, } Ef_t = \frac{\text{I.S.R.}}{\text{A.S.R.}} \times 100$$

$$= \frac{7.35}{9.86} \times 100$$

$$= 74.5\%$$

As previously defined, the total steam system energy input is calculated by the following equation.

$$Q_t = \frac{f_s \times (h_i - h_r)}{1\ 000\ 000}$$

Refer to Table 6 and interpolate the h_f column for 100°C condensate.

$h_r = 419$ kJ/kg.

$$\text{Total steam system energy input, } Q_t = \frac{3313 \times (3211 - 419)}{1\ 000\ 000}$$

$$= 9.25 \text{ GJ/h}$$

As previously defined, the energy extracted from the steam is calculated by the following equation.

$$\text{Energy extracted, } Q_e = \frac{(f_s \times h_i) - (f_{co} \times h_{fo}) - [(f_s - f_{co}) \times h_{go}]}{1\ 000\ 000}$$

Refer to Table 6 and interpolate the h_f and h_g columns for 153°C steam.

Enthalpy of outlet condensate, $h_{fo} = 645$ kJ/kg.

Similarly, enthalpy of outlet steam, $h_{go} = 2749$ kJ/kg

$$\text{Therefore, } Q_e = \frac{(3313 \times 3211) - (99 \times 645) - [(3313 - 99) \times 2749]}{1\,000\,000}$$

$$= 1.74 \text{ GJ/h}$$

Since the water heating system condenses all of the exhaust steam, the steam flow rate attributable to the turbine (f_{st}) can be calculated by the following equation.

$$f_{st} = f_s \times \frac{Q_e}{Q_t}$$

$$= 3313 \times \frac{1.74}{9.25}$$

$$= 623 \text{ kg/h}$$

The annual cost of steam attributable to the turbine, can be calculated.

$$\text{Annual steam cost} = f_{st} \times h \times Cs$$

$$= 623 \times 8400 \times 0.022$$

$$= \$115{,}130/\text{yr}$$

The annual value of electrical energy produced by the alternator can be calculated by the following equation.

$$\text{Annual value of electrical energy} = Wa_o \times Ce \times h$$

$$= 309 \times 0.05 \times 8400$$

$$= \$129{,}780$$

The turbine/alternator set therefore provides a net annual energy cost saving.

$$\text{Annual energy cost saving} = \$129{,}780 - \$115{,}130$$

$$= \$14{,}650/\text{yr}$$

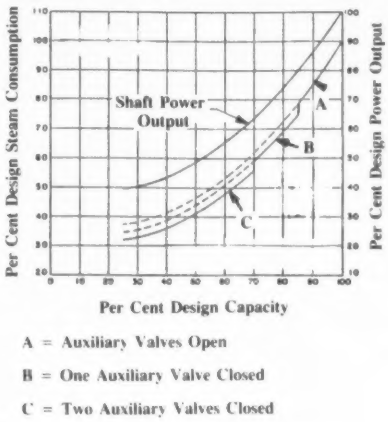
Steam Turbine Control

An important aspect of a turbine application is the method of controlling, or *governing* the turbine speed and power output to match the requirements of the driven device. A wide variety of governing systems are available to meet the requirements of particular applications. A complete discussion of their characteristics is beyond the scope of this module, however, the fundamental purposes and performance effects are outlined. The purpose of a turbine governing system usually serves one of two basic functions.

- Maintain constant turbine speed in spite of fluctuations in load and inlet steam pressure.
- Vary the turbine speed to suit a varying requirement of the driven device.

Table 7 includes a tabulation of the effect on steam rate of power and speed.

The simplest form of turbine control consists of a single governor controlled steam admission valve, perhaps augmented by one or more auxiliary valves, usually manually operated. The auxiliary valves are used to close off nozzles supplying the turbine steam chest (Figure 23) under part load conditions. Figure 25 shows the effect of auxiliary valves on part load turbine performance.



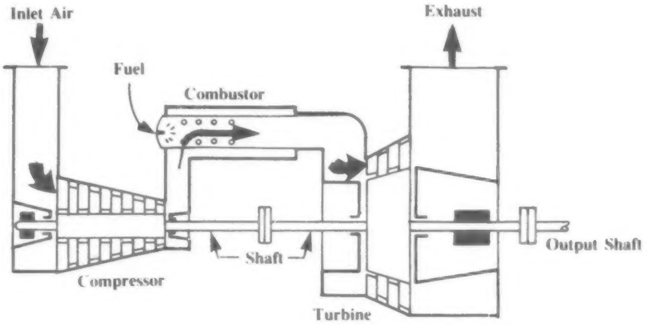
Typical Part Load Performance With Auxiliary Valves
Figure 25

Gas Turbine Types

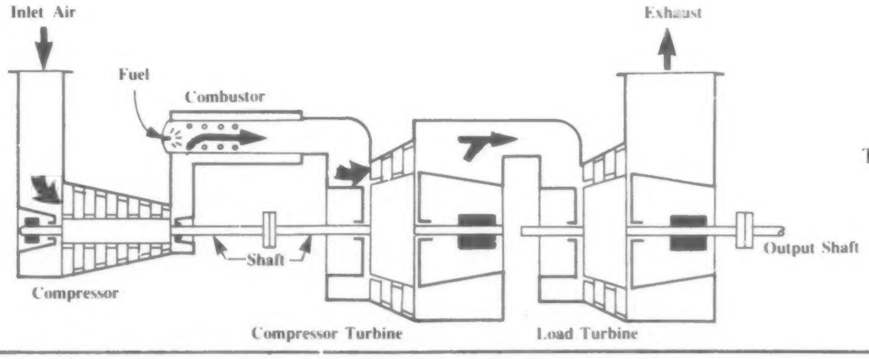
A gas turbine compresses combustion air, burns fuel in a combustor and directs the resulting hot gases through a series of turbine blades. The compressor is usually of the centrifugal or axial type, and is driven by the turbine. Gas turbines are classified as single shaft or two shaft units.

In a *single shaft* gas turbine (Figure 26) a single turbine drives both the compressor and the output shaft. In a *two shaft* gas turbine (Figure 27) a separate turbine drives the output shaft.

The scope of this module is limited to turbines which draw their intake air from the atmosphere (open cycle), derive all their input energy from the fuel supplied to the combustor, and produce an excess of shaft power over that consumed by the compressor.


Inlet Air
Exhaust
Fuel
Combustor
Shaft
Output Shaft
Compressor
Turbine

Single Shaft Gas Turbine
Figure 26


Inlet Air
Exhaust
Fuel
Combustor
Shaft
Output Shaft
Compressor
Compressor Turbine
Load Turbine

Two Shaft Gas Turbine
Figure 27

Gas Turbine Operation

The purpose of gas turbines is to produce either shaft power, a high velocity and high temperature exhaust gas stream, or a combination of both. The shaft power may be used to drive a compressor, pump or fan, and the high temperature exhaust gas may be used to produce steam or hot water in a heat recovery boiler. Gas turbines may also be used as supplementary parts of combined systems where energy from other sources may be used to add to the input to the turbine, and where the exhaust gas may be used as hot combustion air in another burner. The analysis of such integrated systems is beyond the scope of this module.

The *combustor* can burn a variety of fuels, including natural gas, diesel fuel and heavy residual oils. The compressed air is used to support the combustion process and to cool various parts of the gas turbine. Compressed air that has not been taken into the combustor chamber surrounds and cools the combustor shell. Some of this air is emitted through fine slots over the internal surface of the combustor forming a cool film of air to protect the shell from the high gas temperatures. Prior to entry into the turbine section, the combustion gases must be mixed with some diverted air so as not to exceed the temperature limits of the turbine section components. The annular space around the shaft of the turbine section is also cooled by a portion of the diverted air from the compressor.

The turbine section converts the high velocity of the expanding gas to shaft power. The blading is similar to that of steam turbines for impulse and reaction characteristics but the materials must withstand the 750 to 1000°C temperature of the tempered combustion gas.

To achieve optimum combustion and turbine cycle efficiency the combustion air is compressed to about 1200 kPa(gauge). This consumes approximately 66 per cent of the total shaft power.

Gas Turbine Performance Measurements

To evaluate a gas turbine, certain measurements must be taken and specific information must be obtained. For reliable results, the fuel flow, load and temperature should be held constant during measurement.

The *fuel flow rate* can be determined from bulk measurement of tank levels at the beginning and end of a test period, or from a totalized fuel recorder reading. Testing of gas turbines during fluctuating loads requires instantaneous fuel flow readings.

At a formal testing facility the *shaft output* may be measured directly by a dynamometer. For installations driving compressors, fans or pumps, the most appropriate method is to measure the output of the driven device and apply the fundamentals applicable to compressors from this module and the fundamentals applicable to fans and pumps from Module 13. When the turbine is driving electrical generation equipment, output measurements can be taken and applied to the fundamentals for alternators as discussed in Appendix E.

Gas Turbine Performance

Manufacturers' performance information is the most reliable source of data for gas turbine operating characteristics. Because of the number of variables affecting performance, the data usually must be obtained from the manufacturer for each application. Published catalogues usually list only maximum potential shaft power outputs for particular standard models. When manufacturers' data is not available, the energy consumption of a particular gas turbine can be estimated using measured data and the equations that are discussed in this section.

Gas Turbine Performance Characteristics

The nameplate shaft power output of a gas turbine is usually expressed in kilowatts or horsepower, and includes any gearbox loss. Unless the gas turbine is especially designed for a particular application, the nameplate shaft power output is normally given for standard conditions of intake air at 15°C (288.15 K), sea level altitude, and static pressures at inlet and exhaust flanges of 101.325 kpa(absolute). The following are correction factors for gas turbine shaft power output owing to variations from the standard conditions.

- *Inlet temperature*. Each 10 Kelvin units rise will decrease power output by 9 per cent.
- *Altitude*. Each 100 metre increase will reduce power output by 1.15 per cent.
- *Inlet pressure*. Each 10 pascal drop will reduce power output by 0.2 per cent.
- *Outlet pressure*. Each 10 pascal increase will reduce power output by 0.12 per cent.

The pressures at the inlet and exhaust flanges are affected by ancillary equipment such as intake filters, silencers, and waste heat boilers.

The corrected value of the turbine nameplate shaft power output must account for variations from the standard conditions for a particular installation. By combining the constants, the corrected output can be estimated by the following equation.

$$Wt_{oc} = Wt_{on} \times \left[1 - \frac{(T_i - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri}}{5000} - \frac{P_o}{8333} \right]$$

Where, Wt_{oc} = corrected turbine shaft power output (kW)

Wt_{on} = nameplate turbine shaft power output (kW)

T_i = inlet temperature (K)

AL = altitude (m)

P_{ri} = reduction of static pressure below atmospheric at inlet flange (Pa)

P_o = increase of static pressure above atmospheric at outlet flange (Pa)

Constants = combined correction factors for variations from standard conditions as previously discussed.

The *load* on a gas turbine is the ratio of the shaft power output (Wt_o) to the corrected nameplate output as expressed by the following equation.

$$\text{Load} = \frac{Wt_o}{Wt_{oc}}$$

Gas Turbine Efficiency

The performance or *thermal efficiency* of a gas turbine is defined as the energy output divided by the total energy supplied. Thermal efficiency can be expressed by the following equation.

$$Ef_g = \frac{Wt_o \times 3.6}{Q_g} \times 100$$

Where, Ef_g = gas turbine thermal efficiency (%)

Q_g = energy input to the gas turbine (MJ/h)

3.6 = conversion from KW to MJ/h

100 = conversion from decimal to per cent

The total energy supplied to a gas turbine is the higher heating value of the fuel input. The energy input to the gas turbine (Q_g) can be approximated by the following quotation.

$$Q_g = f_f \times HHV \times 3600$$

Where, f_f = fuel flow rate (L/s for liquid fuel or m^3/s for gaseous fuel)

HHV = fuel higher heating value (MJ/L for liquid fuel or MJ/m^3 for gaseous fuel) (supplier data or Appendix C).

3600 = conversion from seconds to hours

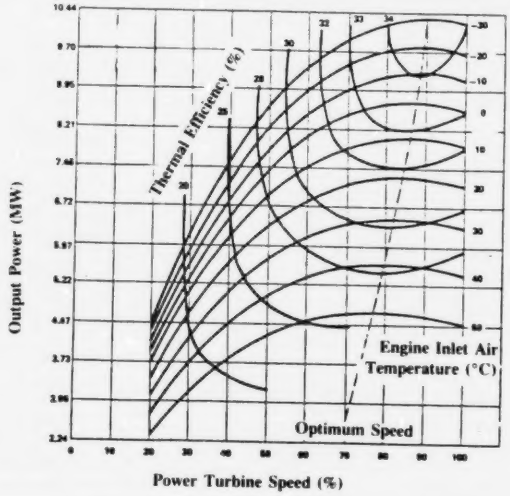
The annual fuel cost ($/yr) to operate the turbine can be calculated by the following equation.

Annual fuel cost = $f_f \times Cf \times h \times 3600$

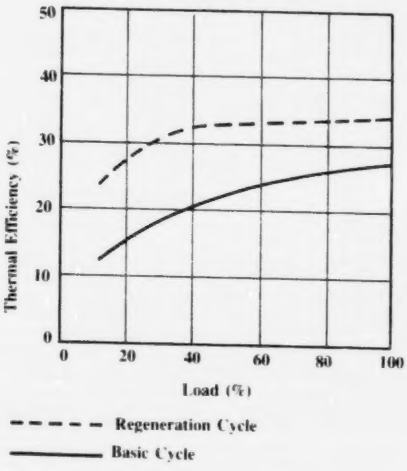
Where, Cf = unit fuel cost ($/L for liquid fuel or $\$/m^3$ for gaseous fuel).

Gas Turbine Energy Consumption

A typical gas turbine, such as that of Figure 26 or 27, would have a thermal efficiency of 18 to 36 per cent. For example, the eight megawatt turbine engine represented in Figure 28 has a maximum thermal efficiency of just over 34 per cent at a compressor air inlet temperature of −30°C (243.15K). The shaft power required to drive the compressor and the energy discharged in the exhaust gas result in an overall thermal efficiency similar to other combustion cycle engines. Measures which increase the efficiency of the components or reduce the energy discharged in the exhaust gas will improve the thermal efficiency.



Basic Gas Turbine Performance Characteristics For An 8 MW Turbine
Figure 28



Part Load Efficiency Of Gas Turbines
Figure 29

One method of improving thermal efficiency is the addition of a *regenerator*. This unit is a heat exchanger that preheats the combustion air between the compressor and the combustor using reject heat from the turbine exhaust gas. Use of a regenerator may raise the overall thermal efficiency to 38 per cent. A regenerator also maintains the thermal efficiency of the gas turbine over a greater power output range, thus providing improved part-load efficiency. Figure 29 shows the effect on efficiency versus per cent load for a typical regenerator installation.

The amount of energy in the exhaust gas also represents a major opportunity for energy recovery which can reduce the net cost of operating the turbine. The temperature of the exhaust gases permits their use for generating low pressure steam, hot water, or for indirect air heating.

The *rating* of a gas turbine is normally presented in fuel consumption per unit of shaft output power, or *specific fuel consumption* in megajoules per kilowatt hour (MJ/kWh). This rating is a means of comparing different gas turbines and evaluating changes to a particular installation. It does not account for any exhaust gas heat recovery that might be incorporated. Since the nameplate rating is usually published for the same standard conditions as the power output, the corrected rating of a particular installation based on the published rating can be calculated by the following equation.

$$R_c = R_n \times \frac{Wt_{oc}}{Wt_{on}}$$

Where, R_c = corrected turbine rating (MJ/kWh)

R_n = nameplate turbine rating (MJ/kWh)

Wt_{oc} = corrected turbine shaft power output (kW)

Wt_{on} = nameplate turbine shaft power output (kW)

The corrected rating normally represents the maximum output at optimum speed. The actual rating of a gas turbine installation operating at a particular load and speed may be estimated by the following equation.

$$R = \frac{Q_g}{Wt_o}$$

Where, R = actual rating (MJ/kWh)

Q_g = energy input to the gas turbine (MJ/h)

Wt_o = turbine shaft power output (kW)

Or, using an actual fuel flow rate, the following equation applies.

$$R = \frac{f_f \times HHV \times 3600}{Wt_o}$$

Where, f_f = fuel flow rate (L/s for liquid fuel or m^3/s for gaseous fuel)

HHV = fuel higher heating value (MJ/L for liquid fuel or MJ/m^3 for gaseous fuel) (Supplier data or Appendix C)

3600 = seconds per hour

The effect on fuel consumption for a particular output power owing to a change in inlet or outlet conditions can be estimated from the inverse ratio of the corrected outputs.

$$f_{f2} = f_{f1} \times \frac{Wt_{oc1}}{Wt_{oc2}}$$

Where, f_{f1}, f_{f2} = initial and revised fuel flow rates (L/s or m^3/s)

Wt_{oc1}, Wt_{oc2} = initial and revised corrected turbine shaft power outputs (kW)

The equation can also be stated in terms of thermal efficiency.

$$f_{f2} = f_{f1} \times \frac{Ef_{g1}}{Ef_{g2}}$$

Where, Ef_{g1}, Ef_{g2} = initial and revised gas turbine thermal efficiencies (%)

The annual fuel cost saving ($/yr) can then be calculated.

Annual fuel cost saving = $(f_{f1} - f_{f2}) \times h \times Cf \times 3600$

Where, h = operation time (h/yr)

Cf = unit fuel cost ($/L or $/m^3)

3600 = seconds per hour

Gas turbines lose heat energy to the surroundings, to lubrication systems and to the exhaust gases. A small amount, which is usually not recoverable, is lost through radiation. The heat energy given up to the lubrication system and through the exhaust gases is significant and is often recovered by auxiliary systems. Air or water heat exchangers are used to cool the lubricating oil. The amount of energy given up through lubricating oil cooling can be estimated by the following equation.

$$Q_o = Q_g \times \textit{factor}$$

Where, Q_o = energy to oil cooling (kJ/h)

Q_g = energy input to the gas turbine (MJ/h)

factor = 4.44 for a basic cycle turbine or 6.55 for a turbine with a regenerator, which includes conversion from MJ to kJ.

The exhaust gas from a turbine contains a large amount of heat energy. Comparative evaluations indicate the heat content above 150°C to be about 52 per cent of the energy input to the gas turbine.

$$Q_x = Q_g \times 0.52$$

Where, Q_x = energy lost in exhaust (MJ/h)

0.52 = decimal portion of input energy

Gas Turbine Performance Analysis

An example performance analysis will be carried out on a gas turbine that is used to drive an air compressor that requires a shaft power input of 3005 kW.

From nameplate information and measurements, the following data was compiled.

Altitude, AL	920 m
Fuel type	natural gas
Fuel flow rate, f_f	0.41 m^3/s
Fuel higher heating value, HHV	37.2 MJ/m^3 (Appendix C)
Unit fuel cost, Cf	$0.21/m^3

Nameplate turbine shaft power output, Wt_{on}	3169 kW
Required turbine shaft power output, Wt_o	3005 kW (air compressor power input)
Nameplate turbine rating, R_n	20.80 MJ/kWh
Operation time, h	7900 h/yr
Inlet temperature, T_i	10°C (283.15K)
Reduction of static pressure below atmospheric at inlet flange, P_{ri}	60 Pa
Increase of static pressure above atmospheric at outlet flange, P_o	750 Pa

Energy input to the gas turbine, $Q_g = f_f \text{ x HHV x } 3600$

$= 0.41 \text{ x } 37.2 \text{ x } 3600$

$= 54\ 907 \text{ MJ/h}$

Annual fuel cost $= f_f \text{ x Cf x h x } 3600$

$= 0.41 \text{ x } 0.21 \text{ x } 7900 \text{ x } 3600$

= \$2,448,684/yr

Energy to oil cooling, $Q_o = Q_g \text{ x } 4.44$

$= 54\ 907 \text{ x } 4.44$

$= 244 \text{ kJ/h}$

Energy lost in exhaust, $Q_x = Q_g \text{ x } 0.52$

$= 54\ 907 \text{ x } 0.52$

$= 28\ 552 \text{ MJ/h}$

The findings of this analysis could be used as a basis for comparing various alterations to the operating conditions to make maximum use of the fuel energy. The energy required to cool the oil may be used to heat water or air for other processes, and the exhaust heat energy may be recovered in a waste heat boiler.

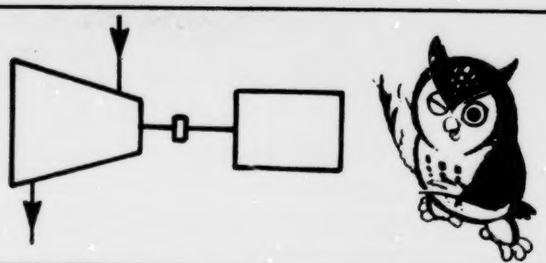
Summary

Numerous energy and cost saving opportunities exist for turbines. Alert personnel, with an awareness of energy management techniques, can easily learn to recognize these opportunities and benefit from them.

Turbines are efficient driving devices where the outlet heat energy can be effectively used for other purposes.

Turbines can be analyzed to estimate the energy input, energy transfer, losses, costs and potential savings. Worksheets 14-T1 through 14-T6 summarize the data and calculations necessary for analysis of turbines and their auxiliaries.

EQUIPMENT SYSTEMS



Steam Turbines

Steam condensing and noncondensing turbines are classified into further configuration subdivisions (Figure 19) as follows.

- Reheat
- Straight flow
- Extraction
- Induction

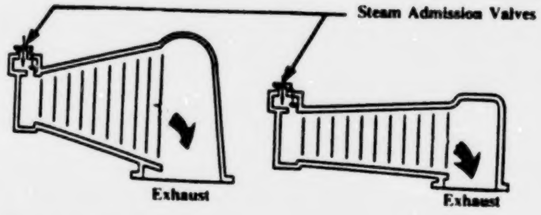
Table 8 lists typical capacity characteristics of each configuration.

Steam Turbine Configurations

In a *reheat turbine* the steam flows through one stage and is extracted, reheated and returned to the turbine at the next lower stage for further expansion. Turbines of this type are normally used in power plants.

Straight flow turbines (Figure 30) use regulated steam flow from inlet to exhaust. These turbines are the simplest in operation since they do not have any complicated interstage valving arrangements. Straight flow turbines are typically used to drive fans and pumps. Straight flow turbines have a low first cost and good application flexibility.

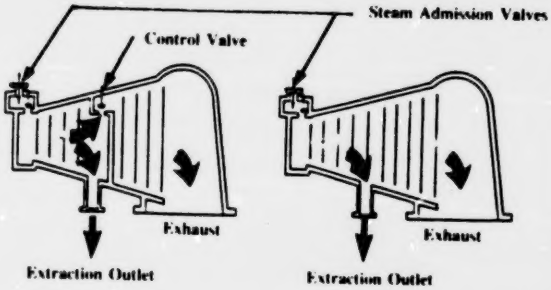
An *extraction turbine* (Figure 31) allows steam to be bled off automatically or at a constant rate at various points along the turbine for use in processes, heating or boiler feedwater heating. By regulation of the bleed off rates, speed and power output of the turbine can be regulated. When additional shaft power is required, the amount of steam extracted can be reduced. Such turbines, operating in industrial and power plants, may serve a dual role of reducing steam pressure for process use while producing shaft power for driving electrical generators or other equipment.



Straight Condensing

Straight Noncondensing

Straight Flow Steam Turbines
Figure 30

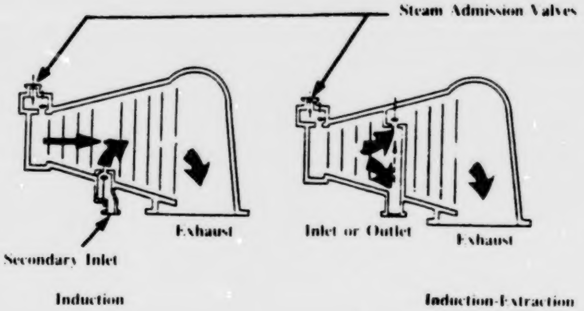


Automatic Extraction

Nonautomatic Extraction

Extraction Steam Turbines
Figure 31

An *induction turbine* (Figure 32) has intermediate steam connections similar to an extraction turbine, but reduced pressure steam is supplied to the turbine at a stage downstream of the main throttle valve to supply a portion of the total input. Combinations of the induction and extraction processes may occur on the same turbine through separate ports. The examples in Figure 32 show only one intermediate connection but additional ports may be used for induction and extraction. This arrangement allows optimum use of energy in the steam to suit the process requirements and achieve maximum overall system efficiency.


Steam Admission Valves
Exhaust
Secondary Inlet
Induction
Inlet or Outlet
Exhaust
Induction-Extraction

Induction Steam Turbines
Figure 32

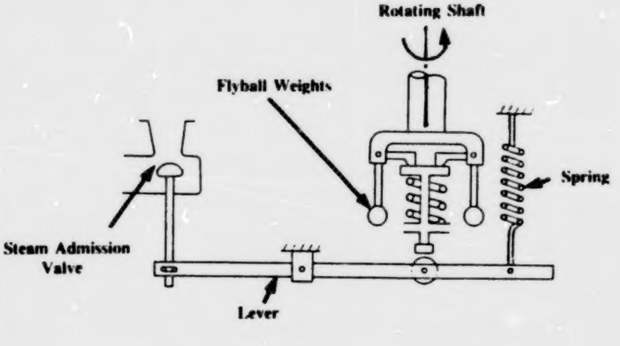
Steam Turbine Control Devices

As discussed in the Fundamentals section, steam turbine controllers, or governors, serve one of two basic functions.

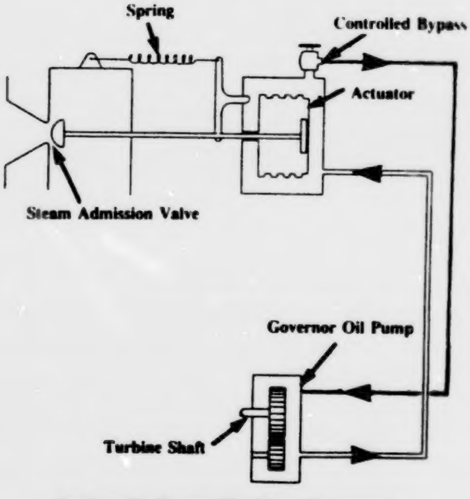
- Maintain constant turbine speed with varying load
- Vary turbine speed to suit a varying load requirement.

The simplest form of constant speed governor is the direct acting flyball type (Figure 33) which uses centrifugal force through a mechanical linkage to adjust the steam admission valve. It is used on single stage mechanical drive turbines with speeds up to 5000 rpm and steam pressures up to 4150 kPa(gauge). For more precise constant or variable speed control, the oil relay type of automatic governor (Figure 34) uses speed induced oil pressure changes, amplified by an actuating device, to adjust the steam admission valve. A large turbine might utilize multiple actuators and steam admission valves combined with devices to vary the speed setting. Manually operated auxiliary steam valves are sometimes provided downstream of the steam admission valve to restrict steam flow to some of the turbine nozzles and permit efficient governor control for reduced load conditions.

In addition to speed governing controls, certain safety devices are used to prevent overspeed and to shut down the turbine upon a signal from protective controls on the driven device. The safety devices commonly include a quick tripping valve, independent of the main steam admission valve, to shut off the steam supply to the turbine.


Rotating Shaft
Flyball Weights
Spring
Steam Admission Valve
Lever

Flyball Type Steam Turbine Governor
Figure 33


Spring
Controlled Bypass
Actuator
Steam Admission Valve
Governor Oil Pump
Turbine Shaft

Hydraulic Steam Turbine Governor
Figure 34

Application of Steam Turbines

Steam turbines have particular applications in industrial, commercial and institutional facilities where the use of steam to produce mechanical power can be economically combined with other uses of the energy in the steam. For plants with existing steam supplies, turbines can often be an efficient means of driving large equipment with high power demands. Examples include fans in the range of 75 to 1900 kW, pumps in the range of 18 to 1900 kW, industrial generators from 10 to 50 000 kW, compressors from 75 to 7500 kW, and utility power plant generators up to 800 MW.

A specific application of straight through condensing turbines is for driving compressors on large heat pumps and air conditioning chillers. The exhaust may be directed to a steam condenser or to an absorption chiller to take advantage of the energy in the discharge steam. Additional details on this and other applications of steam turbines may be found in the Fundamentals and Equipment Handbooks of the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE).

Advantages/Disadvantages of Steam Turbines

In plants with suitable steam supplies, steam turbines may be more economical than large electric motors or gas turbines. Steam turbines have higher output than gas turbines of similar size and energy consumption. Through a cogeneration agreement with the electrical utility, turbine driven electric alternators can be used to provide a portion of the electric power requirement of a process plant during peak loads and supply power to the utility at other times. This arrangement achieves a high utilization of the turbine and steam generation systems, and can provide an acceptable payback of the capital cost if appropriate rates are available for primary energy and the sale of the spare power generated. Steam turbines are available in a wide range of sizes and configurations. Certain designs have common shafts with pumps for industrial and utility applications.

Disadvantages of steam turbines include the need to be permanently associated with a reliable steam source, the requirement for high quality steam with associated feedwater quality control, potential for damage by impurities in the steam (including water), and the requirement for special startup procedures.

Gas Turbines

As identified in Figure 19, gas turbines are available in two basic configurations: *single shaft* and *two shaft*. Both basic configurations are available with many refinements and auxiliary devices that are beyond the scope of this discussion. Gas turbines are available in a range of sizes from a few kilowatts to 140 000 kilowatts. Turbines larger than 2000 kW normally have shaft speeds of 3000 to 6000 rpm. Smaller units may operate at speeds in excess of 20 000 rpm.

Gas Turbine Configurations

A typical *single shaft* gas turbine is shown in Figure 26. The shaft carries the power produced in the turbine to both the compressor and the load. A gearbox may be used in some applications to permit the output shaft speed to be matched to the requirements of the load.

Starting a single shaft gas turbine requires that it be driven by a starting motor at a speed sufficient to compress the combustion air. Since this speed approaches the operating range and all equipment is connected to a single shaft, the starting motor requires a high power input. At some installations the driven load may be disconnected from the turbine shaft during starting, by a coupling device. Single shaft turbines are used where frequent starting is not required, such as for large power generation units.

A *two shaft* gas turbine (Figure 27) has one shaft connecting the compressor and the first turbine, often referred to as the gas producer or compressor turbine, and a second shaft in the power output turbine. The first turbine produces enough power to run the compressor, and the second turbine, which may be referred to as the power turbine or free turbine, drives the output shaft.

Starting a two shaft gas turbine requires only the compressor and the first turbine to be brought up to speed. This reduces the size of the starting equipment, the power required for starting, and the complexity of the complete drive train. As the first turbine approaches full speed under its own power the second turbine is started by the flow of combustion gas.

Gas Turbine Auxiliaries

Equipment used to increase the efficiency of a gas turbine include regenerators and intercoolers.

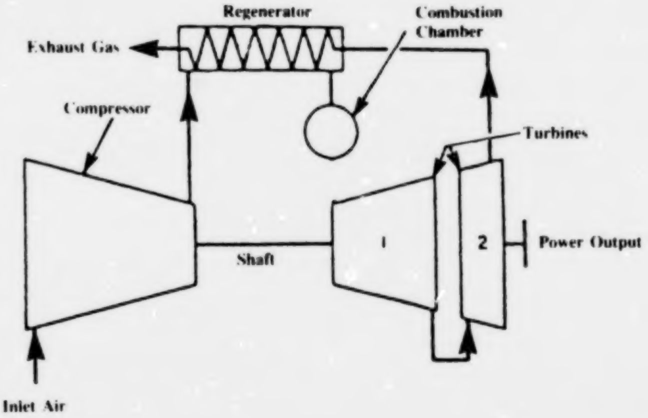
A *regenerator* (Figure 35), sometimes referred to as a recuperator, is a heat exchanger that extracts heat from the exhaust gas of the turbine and preheats the combustion air at the inlet to the combustor. The addition of a regenerator is more easily accomplished on a larger turbine which provides a better opportunity to divert the air through the regenerator.

An *intercooler* is a heat exchanger that cools the air between stages of the compressor. This device is used on turbines having two or more compressors operating in series.

The installation of regenerators or intercoolers on existing gas turbines requires expert assistance. Uniform air flow must be maintained at the combustion chambers and compressor stage inlets to avoid unbalanced, uneven cooling and hot spots. The correct installation of these devices can markedly improve efficiency and extend the performance range of the turbine.

Gear boxes for speed reductions or increases are normally supplied as an integral part of a turbine. Their components must be appropriately matched to the speed and load characteristics of the turbine.

Except for very large installations, turbine driven alternators for electric power generation are also often selected and matched to the turbine characteristics by the turbine supplier.



Gas Turbine Regenerator
Figure 35

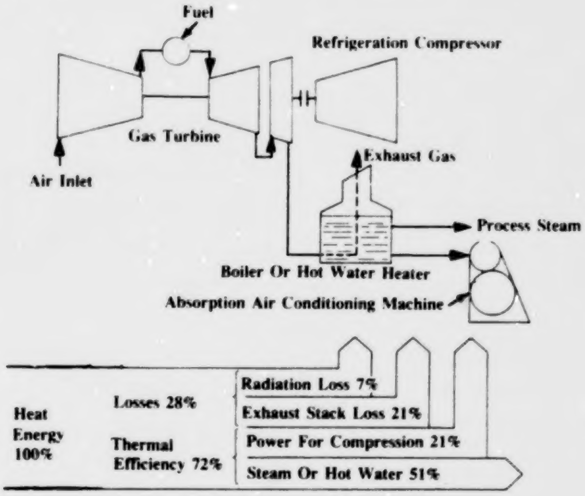
Advantages/Disadvantages of Gas Turbines

Gas turbines have diverse applications. All that is necessary for their operation is a fuel supply, a means of starting and an exhaust system. Compared to other internal combustion engines the gas turbine offers several advantages.

- No external cooling required.
- A variety of fuels can be used.
- Quick starting.
- Small in size with a high power to weight ratio.
- Low vibration.
- Low pollution exhaust.
- High reliability and low maintenance.
- Low capital cost.

Gas turbines can serve as easily installed driving devices for unattended operation in remote areas. They are used in integrated systems (Figure 36) in which the exhaust gas heat energy is utilized. Such systems sometimes comprise the *total energy system* for a facility. The overall thermal efficiency of such systems may exceed 95 per cent.

Disadvantages of gas turbines include low individual thermal efficiency, the large volume of hot exhaust, the need for an auxiliary starting system, and high noise level.


Fuel
Refrigeration Compressor
Gas Turbine
Exhaust Gas
Air Inlet
Process Steam
Boiler Or Hot Water Heater
Absorption Air Conditioning Machine
Heat Energy 100%
Losses 28%
Thermal Efficiency 72%
Radiation Loss 7%
Exhaust Stack Loss 21%
Power For Compression 21%
Steam Or Hot Water 51%

Gas Turbine System With Integrated Heat Recovery
Figure 36

ENERGY MANAGEMENT OPPORTUNITIES

Energy Management Opportunities is the term that represents the ways that energy can be used wisely to save money. A number of Energy Management Opportunities, subdivided into Housekeeping, Low Cost, and Retrofit categories are outlined in this section, with worked examples or text to illustrate the potential energy savings. This is not a complete listing of the opportunities available for steam and gas turbines. However, it is intended to provide ideas for management, operating, and maintenance personnel to allow them to identify other opportunities that are applicable to a particular facility. Other modules in this series should be considered for Energy Management Opportunities applicable to other types of equipment and systems.

Housekeeping Opportunities

Implemented housekeeping opportunities are energy management actions that are done on a regular basis and never less than once a year. The following are typical Energy Management Opportunities in this category.

1. Clean or replace air intake filters.
2. Regularly check for vibration.
3. Ensure that steam turbines are operated at optimum steam and condensate conditions.
4. Ensure that gas turbines are operated at optimum inlet and outlet conditions.
5. Verify and maintain clearance tolerances at turbine rotating elements and seals.
6. Shut down turbine driven equipment when low loads make operation uneconomical.
7. Refine and implement appropriate maintenance measures.
8. Ensure that all speed control systems are functioning properly.

Housekeeping Worked Examples

The following examples, numbered to correspond with the previously listed opportunities, illustrate situations where energy can be saved.

1. Clean or Replace Air Intake Filters

Clean and efficient intake air filters are essential to the reliability of a gas turbine. Dust and other impurities entering the turbine can cause fouling of compressor blades, buildup in passages, and excessive wear at bearings, seals and other mating surfaces.

Dirty filters restrict the flow of inlet air to the turbine which reduces its overall output. By cleaning or replacing the intake filters on a regular basis, the inlet pressure drop is reduced and energy cost savings can be achieved.

2. Check for Vibration

Abnormal vibration is an indication of mechanical problems. Vibration can be caused within a machine by faulty bearings, mechanical imbalance or dynamic imbalance of gases or liquids. Vibration in electric generators and alternators can also be caused by electric circuit problems. These sources of vibration may cause significant energy loss, and should therefore be monitored and corrected as quickly as possible.

3. Operate Steam Turbines at Optimum Conditions

Steam should be supplied at the maximum possible enthalpy (h_i) and exhausted at the lowest possible enthalpy (h_o). The greater the difference between h_i and h_o, the greater the output. On noncondensing turbines the exhaust pressure should be kept at the lowest pressure acceptable to the low pressure steam system.

4. Operate Gas Turbines at Optimum Conditions

Gas turbines should be operated with the least inlet suction loss and the lowest exhaust back pressure practical. Periodic checks should be carried out to ensure that inlet and exhaust passages are kept free of obstructions.

5. Check and Maintain Turbine Clearances

Turbines, and associated gas turbine compressors have close tolerances to minimize leakage and flow deflection. These clearances should be measured periodically and compared with manufacturers' recommendations and past records. Such comparisons will reveal potential areas of leakage and lost efficiency.

6. Shut Systems Down

Figure 25 indicates that, between zero and 25 per cent output, the particular steam turbine uses a constant 32 per cent of the design steam consumption.

Gas turbines must be operated at approximately 50 per cent of full speed to maintain the compression cycle. This results in significant energy cost to operate the compressor at idle with no output or for a very low load.

Shutting down turbines can save significant energy cost when low loads can be handled by alternate means such as electric motors.

7. Maintenance Program

An appropriate maintenance program for turbines should be tailored to the specific needs of the facility. Maintenance programs should typically include the following items.

- Daily: Observe turbine sound, level of vibration, lubricating oil temperature, various seals and connections, leaks, and auxiliary equipment. Read installed gauges and meters, and input and output instrumentation and recordings.
- Monthly: Examine air intake and exhaust systems, test lubricating oils for lubricity and contamination, grease appropriate fittings, test safety systems, change oil filters, examine shaft couplings, examine ignition system and spark plugs, check tightness of all exposed connections and bolts, and check condition of all auxiliary equipment.
- Annually: Examine the fuel nozzles, calibrate all instrumentation, conduct a turbine performance test and perform indicated repairs, thoroughly test all auxiliary equipment, and examine all readily accessible bearings.

8. Verify/Calibrate Speed Controls

The speed control of a turbine may be the most important aspect of its proper operation. Controls may provide for fixed speed operation under fluctuating loads, or varying speed in response to a control function. In all instances the speed control should be checked periodically to ensure that it is functioning in a manner that suits the characteristics of the driven machine and the load profile. In certain applications the speed control is critical to the efficient operation of the turbine driven device. A centrifugal compressor driven by a turbine is one application that depends upon accurate speed control to optimize compressor efficiency under varying load.

Low Cost Opportunities

Implemented low cost opportunities are energy management actions that are done once and for which the cost is not considered great. The following are typical Energy Management Opportunities in this category.

1. Modify or relocate air intake to provide cool air to gas turbines.
2. Recover the heat produced by the oil cooler on a gas turbine.
3. Install optimum insulation on equipment.
4. Add or relocate control components such as temperature and pressure sensors to optimize systems operation.

Low Cost Worked Examples

The worksheets can be used to estimate potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets.

1. Relocate Air Intake

A gas turbine driving an alternator was located in a boiler plant with its air intake from a boiler plant make-up air system at 15°C. It was recognized that turbine fuel cost savings might be achieved by ducting outdoor air directly to the turbine during that period of the year when the outdoor air temperature is below the make-up air system temperature. From weather records it was determined that the outdoor temperature is normally below 15°C for 3200 hours per year and the average outdoor temperature is 5°C for that period. Fuel consumption records indicate an average natural gas fuel consumption for the same period of 0.0424 m^3/s. The unit fuel cost of natural gas was $\$0.21/m^3$.

The following additional data was obtained by instrument measurements and from nameplate information.

Nameplate turbine shaft power output, Wt_{on}	620 kW
Altitude, AL	130 m
Initial reduction of static pressure below atmospheric at inlet flange, P_{ril}	0 Pa(gauge)
Increase of static pressure above atmospheric at outlet flange, P_o	250 Pa(gauge)

It was estimated that the pressure drop in the new duct would create a revised reduction of pressure below atmospheric at the inlet flange of 14 Pa.

Using Worksheet 14-T1, the revised fuel flow rate was calculated to be 0.0389 m^3/s and the annual fuel cost saving was calculated to be $8,467 per year.

The cost to add the intake duct was estimated to be $6,500.

$$\text{Simple payback} = \frac{\$6,500}{\$8,467}$$

$$= 0.8 \text{ years (9 months).}$$

2. Recover Heat from Oil Cooler

A natural gas fired turbine with a regenerator used in a cogeneration application discharged the oil cooling water directly to the sewer. It was recognized that the heat produced by oil cooling could be used to preheat hot water used for plant processes. The turbine used natural gas at a flow rate of 0.75 m^3/s.

The recovered heat would displace saturated steam that has a heating value of 2200 kJ/kg. The unit steam cost was determined by the plant operating staff to be $0.022/kg. The recovered heat is usable 8 hours per day during 260 days per year, or a total of 2080 hours per year.

Using Worksheet 14-T2, the maximum potential annual steam cost saving was calculated to be $1,368 per year. The estimated cost to install a heat recovery heat exchanger and connect the necessary piping is $1,500.

$$\text{Simple payback} = \frac{\$1,500}{\$1,368}$$

$$= 1.1 \text{ years}$$

3. Install Optimum Insulation

Application of insulation around machines such as steam or gas turbines reduces the loss of thermal energy. Thermal energy loss reduces the temperature, and thus the volume and pressure of the steam or combustion gases. Since turbines rely upon the expansion of steam or gas from a higher to a lower pressure, any exterior loss that reduces the pressure will also reduce the energy available within the machine and, consequently, reduce the output of the turbine.

Steam turbines benefit from insulation on all components which carry steam. The condensers on condensing turbines may be insulated to reduce the required amount of space cooling.

The compressor and air ducts of gas turbines can be insulated to retain the heat gained by the air during compression. Regenerators and secondary combustion chambers can also be insulated to retain the heat in the gases and combustion air. When the turbine exhaust is directed to a secondary device, the turbine discharge and ductwork should be insulated to retain the thermal energy in the gas. Even when gas turbines are exhausted to atmosphere, some insulation should be placed on the turbine casing and the discharge duct for personnel safety and to reduce the space cooling requirements.

The application of insulation must not allow the temperature of surfaces and components to exceed manufacturer's limits. Optimum levels of insulation will depend on the desired surface temperature, the surrounding space temperature and the cost of steam and fuel.

4. Optimize the Location of Control Components

The location of control components such as temperature and pressure sensors, flow meters and flow detectors may not be optimum in an existing installation. The removal, relocation or addition of a control device must be performed by personnel experienced with turbines and controls. Prior to performing any work on a turbine the manufacturer should be consulted for discussion of the envisaged change. The manufacturer may provide information on optimizing the changes, proper procedures to follow, measurements to verify the value of the changes, and any detrimental effects to the turbine and auxiliaries.

Retrofit Opportunities

Implemented retrofit opportunities are energy management actions that are done once and for which the cost is significant. Many of the opportunities in this category will require detailed analysis by specialists. Worked examples are provided for some of the listed Energy Management Opportunities, while in other cases there is only commentary. The following are typical Energy Management Opportunities in the retrofit category.

1. Preheat gas turbine combustion air with exhaust gas.
2. Utilize heat from the exhaust of gas turbines.
3. Modify inlet and outlet pipework to reduce flow losses.
4. Utilize the heat from the surface of turbines.
5. Upgrade turbine components for improved efficiency.
6. Optimize controls with most appropriate control devices and systems.
7. Install a back pressure turbine to act as a steam pressure reducing device.
8. Optimize inlet and outlet conditions for steam turbines.

Retrofit Worked Examples

The worksheets can be used to estimate potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets.

1. Regenerator Installation on a Gas Turbine

It was proposed to add a lightweight regenerator to preheat combustion air using the exhaust gas from a natural gas fired turbine in a cogeneration plant. The turbine drives an alternator to produce electric power and continuously produces the alternator's rated output for 351 days, or 8424 hours, per year. A regenerator manufacturer reviewed the turbine specifications and assured the owner a gain of 4 per cent in thermal efficiency at the same output. The unit fuel cost of natural gas was $\$0.21/m^3$.

The following data was measured or compiled from equipment nameplates.

Alternator nameplate power output	900 kW
Rated voltage (alternator nameplate), Vr	12 000 volts
Rated full load current (alternator nameplate), I_r	49.5 amperes
Rated full load power factor (alternator nameplate), $p.f._r$	0.88
Measured voltage (alternator output), V	12 000 volts

Measured current (alternator output), I	49.25 amperes
Measured power factor (alternator output), p.f.	0.88
Phases	3 phase
Initial fuel flow rate, f_{fI}	0.094 m^3/s

Using Worksheet 14-E3, the required alternator shaft power input was calculated to be 978 kW. Since the alternator was direct driven the turbine output power was taken to be equal to the required alternator shaft power input.

Using Worksheet 14-T3, the annual fuel cost saving for a 4 per cent increase in turbine efficiency was calculated to be $76,423 per year.

The regenerator installation is estimated to cost $280,000.

$$\text{Simple payback} = \frac{\$280{,}000}{\$76{,}423}$$

$$= 3.7 \text{ years.}$$

2. Heat Recovery from Gas Turbine Exhaust

The operators of an ore refining plant wished to utilize the exhaust gas from an existing natural gas fired turbine for direct contact drying of ore. The ore was initially dried by direct gas firing and consumed 694 000 m^3 of natural gas per year. The process was operated continuously for 336 days, or 8,064 hours, per year. Based on the measured turbine fuel consumption of 0.106 m^3/s, the turbine manufacturer advised that the exhaust gas could meet the entire drying energy requirement. The estimated back pressure at the turbine exhaust required to move the gas through the dryer is 950 Pa(gauge). The unit fuel cost of natural gas is $0.21/$m^3$.

The following data was measured or recorded from equipment nameplates.

Nameplate turbine shaft power output, Wt_{on}	1170 kW
Altitude, AL	0 m
Initial inlet temperature (average), T_{iI}	15°C (288.15K)
Initial reduction of static pressure below atmospheric at inlet flange, P_{riI}	0 Pa
Initial increase of static pressure above atmospheric at outlet flange, P_{oI}	0 Pa

$$\text{Annual dryer fuel cost saving} = 694\ 000\ m^3/yr \times \$0.21/m^3$$

$$= \$145{,}740/yr.$$

Using Worksheet 14-T1, the fuel cost increase to operate the turbine at the increased exhaust gas pressure was calculated to be $85,349 per year.

The estimated capital cost to install the necessary high temperature ducting and convert the dryer is $220,000.

$$\text{Simple payback} = \frac{\$220{,}000}{\$145{,}700 - \$85{,}349}$$

$$= 3.6 \text{ years.}$$

3. Reduce Connection Loss

The piping and ducts connected to turbines are usually sized to optimize the various constraints of economy, weight, friction loss, space and materials. In many cases friction loss is not the principal constraint. Opportunities may be found to save operating cost by enlarging pipes, ducts, valves, filters, instrumentation fittings, heat exchangers and passageways. However, the manufacturer should be consulted to ensure that any modification will not adversely effect the turbine operation.

4. Utilize Surface Heat from Turbines

Turbines are designed to have some external cooling surfaces because of the high internal temperatures to which they are subjected. The heat emitted from the surfaces may be utilized for space heating, drying processes or preheating process air. The quantity of energy available is best determined by temperature and air flow measurements. Utilization of the heat can yield savings by replacing energy generated by more costly methods.

5. Install Coating on Compressor Blades

During the overhaul of a 12 000 kW gas turbine, replacement compressor blades were coated with iron aluminide to reduce blade surface friction. Records indicate that when the turbine was new it had an output of 12 020 kW at an inlet temperature of 32°C and a natural gas consumption of 1.418 m^3/s. Following the overhaul the same power output and inlet temperature required a gas consumption of 1.331 m^3/s. Coating the blades added \$53,270 to the cost of the retrofit. The cost of fuel was \$0.21/m^3.

Using Worksheet 14-T4, the annual fuel cost saving was calculated to be \$554,063 per year.

$$\text{Simple Payback} = \frac{\$53,270}{\$554,063}$$

$$= 0.1 \text{ years (1 month).}$$

An additional benefit is that the coating is harder than the blade material and is expected to allow extension of the time period between overhauls. Coating of the blades must be done in such a manner that there is no effect on the turbine balance.

6. Install Automated Steam Admission Valves

Automatic auxiliary valves that effectively regulate the number of operating nozzles will significantly improve the efficiency of a steam turbine that operates under varying load conditions. The application of such devices requires expert assistance. Figure 25 in the Fundamentals section illustrates the performance characteristics for a particular turbine with auxiliary valves.

7. Use Turbine for Steam Pressure Reduction

A process plant generates saturated steam at 1825 kPa(absolute) in a boiler. A portion of the steam produced is passed through a pressure reducing valve for use at a pressure of 271 kPa(absolute) for process water heating. It was decided to examine the potential energy cost saving by utilizing a backpressure turbine to effect the pressure reduction and generate electric energy for use in the plant. The steam flow rate for process water heating was measured to be 2500 kg/h during the plant operating period of 16 hours per day, 250 days per year, or 4000 hours per year. For the purpose of this analysis, condensate from the process water heater is assumed to be returned to the boiler at the saturation temperature corresponding to 271 kPa (absolute).

From a review of available turbine/alternator sets and based on the available steam flow rate, a set was selected that was rated to produce 138 kW of electric power with a saturated steam supply of 2437 kg/h at 1825 kPa(absolute). The rated exhaust steam conditions were 271 kPa(absolute) with 6.4 per cent moisture.

The pressure reducing valve was to remain in operation to maintain constant pressure in the lower pressure steam system and to provide peak steam flow requirements.

Using Worksheet 14-T5, the annual cost of steam attributable to the turbine was calculated to be \$20,416 per year. The value of the electric energy produced was calculated to be \$27,600 per year for a net annual energy cost saving of \$7,184.

The cost of installing the turbine is estimated to be \$32,000.

$$\text{Simple payback} = \frac{\$32,000}{\$7,184}$$

$$= 4.5 \text{ years.}$$

8. Refit Steam Turbine System Condenser

A 670 kW condensing steam turbine driving a large process pump was observed to be operating at 14 kPa(absolute) exhaust steam pressure instead of the design exhaust pressure of 6.0 kPa (absolute). The higher exhaust pressure was found to be caused by a badly fouled condenser. It was decided to determine the steam cost savings that could be achieved by refitting the condenser to its original condition. The process pump operated continuously for 337 days, or 8088 hours, per year. Steam flow records indicated an average turbine inlet steam flow rate of 860 kg/h. Superheated steam was supplied to the turbine at 2312 kPa(gauge) and 260°C. Steam was valued at $0.022 per kg.

Using Worksheet 14-T6 the annual steam cost saving was calculated to be $3,737 per year.

The estimated cost to clean and retube the condenser is $15,000.

$$\text{Simple payback} = \frac{\$15,000}{\$3,737}$$

$$= 4.0 \text{ years}$$

Gas Turbine Inlet and Outlet Conditions

Worksheet 14-T1
(Page 1 of 2)

Company: WORKED EXAMPLE #1 Date: 86/08/19

Location: TURBINES - LOW COST By: MBE

Nameplate turbine shaft power output, Wt_{on}	620	kW
Altitude, AL	130	m
Initial inlet temperature, T_{i1}	288.15	K
Revised inlet temperature, T_{i2}	278.15	K
Initial reduction of static pressure below atmospheric at inlet flange, Pr_{i1}	0	Pa
Revised reduction of static pressure below atmospheric at inlet flange, Pr_{i2}	14	Pa
Initial increase of static pressure above atmospheric at outlet flange, P_{o1}	250	Pa
Revised increase of static pressure above atmospheric at outlet flange, P_{o2}	250	Pa
Intial fuel flow rate (measured), f_{f1}	0.0424	~~L/s or~~ m³/s
Unit fuel cost, Cf	0.21	~~$/L or~~ $/m³
Operation time, h	3200	h/yr

Initial corrected turbine shaft power output,

$$Wt_{oc1} = Wt_{on} \times \left[1 - \frac{(T_{i1} - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri1}}{5000} - \frac{P_{o1}}{8333} \right]$$

$$= 620 \times \left[1 - \frac{(288.15 - 288.15)}{111} - \frac{130}{8696} - \frac{0}{5000} - \frac{250}{8333} \right]$$

$= $ 592 kW

Gas Turbine Inlet and Outlet Conditions

Worksheet 14-T1
(Page 2 of 2)

Company: WORKED EXAMPLE #1 Date: 86/08/19

Location: TURBINES - LOW COST By: MBE

Revised corrected turbine shaft power output,

$$Wt_{oc2} = Wt_{on} \times \left[1 - \frac{(T_{i2} - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri2}}{5000} - \frac{P_{o2}}{8333} \right]$$

$$= 620 \times \left[1 - \frac{(278.15 - 288.15)}{111} - \frac{130}{8696} - \frac{14}{5000} - \frac{250}{8333} \right]$$

$$= \underline{\quad 646 \quad} \text{ kW}$$

Revised fuel flow rate, $f_{f2} = f_{f1} \times \frac{Wt_{oc1}}{Wt_{oc2}}$

$$= 0.0424 \times \frac{592}{646}$$

$= \underline{0.0389}$ ~~L/s or~~ m^3/s

Annual fuel ~~cost~~ saving = $(f_{f1} - f_{f2}) \times h \times Cf \times 3600$

$$= (0.0424 - 0.0389) \times 3200 \times 0.21 \times 3600$$

= \$ 8,467 /yr

Gas Turbine Oil Cooler Heat Recovery

Worksheet 14-T2
(Page 1 of 1)

Company: WORKED EXAMPLE #2 Date: 86/08/19
Location: TURBINES - LOW COST By: MBE

Fuel flow rate (measured), f_f 0.075 ~~L/s or~~ m³/s

Oil heat rejection factor (4.44 for basic cycle or 6.55 for regeneration cycle) 6.55

Operation time (that recovered heat usable), h 2080 h/yr

Fuel higher heating value, HHV (Appendix C) 37.2 ~~MJ/L or~~ MJ/m³

Unit steam cost, Cs $ 0.022 /kg

Displaced steam heating value 2200 kJ/kg (1)

Energy input to gas turbine, $Q_g = f_f \times \text{HHV} \times 3600$

$= 0.075 \times 37.2 \times 3600$

$=$ 10 044 MJ/h

Energy to oil cooling, $Q_o = Q_g \times \textit{factor}$

$= 10\,044 \times 6.55$

$=$ 65 788 kJ/h

Steam flow rate displaced, $f_{sd} = \dfrac{Q_o}{(1)}$

$= \dfrac{65\,788}{2200}$

$=$ 29.9 kg/h

Annual steam ~~cost~~ saving, $= f_{sd} \times h \times Cs$

$= 29.9 \times 2080 \times 0.022$

= $ 1,368 /yr

Electric Alternator Performance

Worksheet 14-E3
(Page 1 of 1)

Company: WORKED EXAMPLE #1 (SHT. 1) Date: 86/08/19

Location: TURBINES - RETROFIT By: MBE

Alternator Data (nameplate or measured)

Rated voltage, V_r	12 000	volts
Rated full load current, I_r	49.5	amps
Measured voltage, V	12 000	volts
Measured current, I	49.25	amps
Phase function, Y (1.73 for 3 phase, 1.0 for 1 phase)	1.73	
Nameplate power output	900	kW
Rated full load power factor, $p.f._r$	0.88	(decimal)
Measured power factor, p.f.	0.88	(decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{49.25 \times 12\,000 \times 0.88}{49.5 \times 12\,000 \times 0.88}$$

$$= 0.99$$

Alternator efficiency, Ef_a (Figure E-1) 0.92 (decimal)

$$\text{Alternator power output, } Wa_o = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{12\,000 \times 49.25 \times 1.73 \times 0.88}{1000}$$

$$= 900 \text{ kW}$$

$$\text{Alternator shaft power input, } Wa_i = \frac{Wa_o}{Ef_a} = \frac{900}{0.92} = 978 \text{ kW}$$

$(= W_{t_o})$

Regenerator Installation On A Gas Turbine

Worksheet 14-T3

(Page 1 of 2)

Company: WORKED EXAMPLE #1 (SHT. 2) Date: 86/08/19

Location: TURNBINES - RETROFIT By: MBE

Turbine shaft power output, Wt_o 978 kW (Wa;-WORKSHEET 14-E3)

Initial fuel flow rate, f_{fl} 0.094 ~~L/s or~~ m^3/s

Increase in turbine efficiency with regenerator (manufacturer) 4 % (1)

Operation time, h 8424 h/yr

Fuel higher heat value, HHV (Appendix C) 37.2 ~~MJ/L or~~ MJ/m^3

Unit fuel cost, Cf 0.21 ~~$/L or~~ $/$m^3$

Initial energy input to the gas turbine, $Q_{gl} = f_{fl} \times HHV \times 3600$

$= 0.094 \times 37.2 \times 3600$

$=$ 12 588 MJ/h

Initial gas turbine thermal efficiency, $Ef_{gl} = \dfrac{Wt_o \times 360}{Q_{gl}}$

$= \dfrac{978 \times 360}{12\,588}$

$=$ 28 %

Revised gas turbine thermal efficiency, $Ef_{g2} = Ef_{gl} + (1)$

$= 28 + 4$

$=$ 32 %

Regenerator Installation On A Gas Turbine

Worksheet 14-T3

(Page 2 of 2)

Company: WORKED EXAMPLE #1 (SHT. 3) Date: 86/08/19

Location: TURNBINES - RETROFIT By: MBE

Revised fuel flow rate, $f_{f2} = f_{f1} \times \frac{Ef_{g1}}{Ef_{g2}}$

$$= 0.094 \times \frac{28}{32}$$

= 0.082 ~~L/s or~~ m^3/s

Annual fuel cost saving = $(f_{f1} - f_{f2}) \times h \times Cf \times 3600$

$$= (0.094 - 0.082) \times 8424 \times 0.21 \times 3600$$

= $ 76,423 /yr

Gas Turbine Inlet and Outlet Conditions

Worksheet 14-T1

(Page 1 of 2)

Company: WORKED EXAMPLE #2 Date: 86/08/19

Location: TURNBINES - RETROFIT By: MBE

Nameplate turbine shaft power output, Wt_{on} 1170 kW

Altitude, AL 0 m

Initial inlet temperature, T_{i1} 288.15 K

Revised inlet temperature, T_{i2} 288.15 K (= T_{i1})

Initial reduction of static pressure below atmospheric at inlet flange, Pr_{i1} 0 Pa

Revised reduction of static pressure below atmospheric at inlet flange, Pr_{i2} 0 Pa

Initial increase of static pressure above atmospheric at outlet flange, P_{o1} 0 Pa

Revised increase of static pressure above atmospheric at outlet flange, P_{o2} 950 Pa

Intial fuel flow rate (measured), f_{f1} 0.106 ~~L/s or~~ m³/s

Unit fuel cost, Cf 0.21 ~~$/L or~~ $/m³

Operation time, h 8064 h/yr

Initial corrected turbine shaft power output,

$$Wt_{oc1} = Wt_{on} \times \left[1 - \frac{(T_{i1} - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri1}}{5000} - \frac{P_{o1}}{8333}\right]$$

$$= 1170 \times \left[1 - \frac{(288.15 - 288.15)}{111} - \frac{0}{8696} - \frac{0}{5000} - \frac{0}{8333}\right]$$

= 1170 kW

Gas Turbine Inlet and Outlet Conditions

Worksheet 14-T1
(Page 2 of 2)

Company: WORKED EXAMPLE #2 Date: 86/08/19

Location: TURNBINES - RETROFIT By: MBE

Revised corrected turbine shaft power output,

$$Wt_{oc2} = Wt_{on} \times \left[1 - \frac{(T_{i2} - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri2}}{5000} - \frac{P_{o2}}{8333}\right]$$

$$= 1170 \times \left[1 - \frac{(288.15 - 288.15)}{111} - \frac{0}{8696} - \frac{0}{5000} - \frac{950}{8333}\right]$$

$= 1037$ kW

Revised fuel flow rate, $f_{f2} = f_{f1} \times \frac{Wt_{oc1}}{Wt_{oc2}}$

$$= 0.106 \times \frac{1170}{1037}$$

$= 0.120$ ~~L/s or~~ m^3/s

Annual fuel cost ~~saving~~ $= (f_{f1} - f_{f2}) \times h \times Cf \times 3600$

$= (0.106 - 0.120) \times 8064 \times 0.21 \times 3600$

= $ (-) 85,349 /yr

(COST INCREASE)

Improved Gas Turbine Performance

Worksheet 14-T4

(Page 1 of 1)

Company: WORKED EXAMPLE #5 Date: 86/08/19

Location: TURBINES - RETROFIT By: MBE

TURBINE DATA: (nameplate or measured)

Initial fuel flow rate, f_{f1} 1.418 ~~L/s or~~ m^3/s

Revised fuel flow rate, f_{f2} 1.331 ~~L/s or~~ m^3/s

Unit fuel cost, Cf 0.21 ~~$/L or~~ $/$m^3$

Operation time, h 8424 h/yr

Annual fuel ~~cost~~ saving = $(f_{f1} - f_{f2}) \times h \times Cf \times 3600$

= (1.418 − 1.331) x 8424 x 0.21 x 3600

= $ 554,063 /yr

Steam Turbine As A Pressure Reducing Device

Worksheet 14-T5
(Page 1 of 2)

Company: WORKED EXAMPLE #7 Date: 86/08/19

Location: TURBINES - RETROFIT By: MBE

Steam flow rate, f_s	2437	kg/h	
Inlet steam pressure	1825	kPa(absolute)	
Inlet steam enthalpy, h_i (h_g — Table 6)	2795	kJ/kg	
Outlet steam pressure	271	kPa(absolute)	
Constant moisture per cent in outlet steam (Figure 24)	6.4	%	(1)
Enthalpy of outlet steam, h_{go} (h_g — Table 6)	2720	kJ/kg	
Enthalpy of outlet condensate, h_{fo} (h_f — Table 6)	548	kJ/kg	
Return condensate temperature	N/A	°C	
Enthalpy of return condensate, h_r	548	kJ/kg (= h_{fo} FOR THIS EXAMPLE)	
Alternator power output, Wa_o (Worksheet 14-E3 or manufacturer)	138	kW	
Unit steam cost, Cs	$ 0.022	/kg	
Unit electrical energy cost, Ce	$ 0.05	/kWh	
Operation time, h	4000	h/yr	

Exhaust condensate flow rate, $f_{co} = f_s \times \frac{(1)}{100}$

$$= 2437 \times \frac{6.4}{100}$$

$$= 156 \text{ kg/h}$$

Energy extracted from the steam,

$$Q_e = \frac{(f_s \times hi) - (f_{co} \times h_{fo}) - [(f_s - f_{co}) \times h_{go}]}{1\,000\,000}$$

$$= \frac{(2437 \times 2795) - (156 \times 548) - [(2437 - 156) \times 2720]}{1\,000\,000}$$

$$= 0.522 \text{ GJ/h}$$

Steam Turbine As A Pressure Reducing Device

Worksheet 14-T5
(Page 2 of 2)

Company: WORKED EXAMPLE #7 Date: 86/08/19

Location: TURBINES - RETROFIT By: MBE

Total steam system energy input, $Q_t = \dfrac{f_s \times (h_i - h_r)}{1\,000\,000}$

$$= \frac{2737 \times (2795 - 548)}{1\,000\,000}$$

$$= 5.476 \text{ GJ/h}$$

Steam flow rate attributable to the turbine, $f_{st} = f_s \times \dfrac{Q_e}{Q_t}$

$$= 2437 \times \frac{0.522}{5.476}$$

$$= 232 \text{ kg/h}$$

Note: This is the additional flow rate of inlet steam required to operate the turbine.

Annual steam cost = $f_{st} \times h \times Cs$

= 232 x 4000 x 0.022

= \$ 20,416 /yr (2)

Value of electrical energy produced = $Wa_0 \times Ce \times h$

= 138 x 0.05 x 4000

= \$ 27,600 /yr (3)

Annual energy ~~cost~~ saving = (3) − (2) = 27,600 − 20,416

= \$ 7,184 /yr

Steam Turbine Inlet and Outlet Conditions

Worksheet 14-T6
(Page 1 of 2)

Company: WORKED EXAMPLE #8 Date: 86/08/19

Location: TURNBINES - RETROFIT By: MBE

Initial steam flow rate, f_{s1} 860 kg/h

INLET CONDITIONS:

Initial inlet steam pressure, P_{i1} 2413 kPa(absolute)

Initial inlet steam temperature, T_{i1} 260 °C

Initial enthalpy of inlet steam, h_{i1} (Figure 24) 2919 kJ/kg

Revised inlet steam pressure, P_{i2} N/A kPa(absolute)

Revised inlet steam temperature, T_{i2} N/A °C

Revised enthalpy of inlet steam, h_{i2} ~~(Figure 24)~~ 2919 kJ/kg $(= h_{i1})$

OUTLET CONDITIONS:

Initial outlet steam pressure, P_{o1} 14.0 kPa(absolute)

Initial outlet steam temperature, T_{o1} N/A °C

Initial enthalpy of outlet steam, h_{o1} (Figure 24 or Table 6) 220 kJ/kg

Revised outlet steam pressure, P_{o2} 6.0 kPa(absolute)

Revised outlet steam temperature, T_{o2} N/A °C

Revised enthalpy of outlet steam, h_{o2} (Figure 24 or Table 6) 152 kJ/kg

Operation time, h 8088 h/yr

Unit steam cost, Cs $ 0.022 /kg

Steam Turbine Inlet and Outlet Conditions

Worksheet 14-T6
(Page 2 of 2)

Company: WORKED EXAMPLE #8 Date: 86/08/19

Location: TURBINES - RETROFIT By: MBE

Revised steam flow rate, f_{s2} $= f_{s1} \times \dfrac{h_{i1} - h_{o1}}{h_{i2} - h_{o2}}$

$= 860 \times \dfrac{2919 - 220}{2919 - 152}$

$=$ 839 kg/h

Annual steam ~~cost~~ saving $= (f_{s1} - f_{s2}) \times h \times Cs$

$= (860 - 839) \times 8088 \times 0.022$

= $ 3,737 /yr

APPENDICES

A Glossary
B Tables
C Common Conversions
D Worksheets
E Electric Motor Drives and Alternators

Glossary

Actual Capacity (Compressor) — the quantity of air or gas compressed and delivered by a compressor. It is usually expressed in L/s at intake pressure and temperature.

Absolute Pressure — pressure referred to a perfect vacuum. It is the sum of gauge pressure and atmospheric pressure, and is expressed as kPa(absolute).

Adiabatic Compression — an ideal compression process during which no heat is extracted from or added to the system. The temperature of the air rises steadily during compression, which requires a greater amount of work.

Absolute Temperature — temperature expressed in degrees Kelvin (K). The indicated temperature on the Celsius scale plus 273.15.

Ambient Temperature — the temperature of the medium surrounding a device.

Boyle's Law — at a constant temperature, the volume of a gas varies inversely with the pressure to which it is subjected, $P_1 \times V_1 = P_2 \times V_2$.

Centrifugal Force — the force that tends to impel objects outward from a center of rotation.

Clearance Volume — the volume of the space at the end of the compression cycle in a positive displacement compressor. For a piston type compressor, it would be the volume between the piston and the discharge valve at the end of the compression stroke.

Compression Efficiency — the ratio of work required to compress adiabatically and reversibly all gas delivered by a compressor (per stage) to the actual work delivered to the gas by the piston or blades of the compressor.

Compression Ratio — the numerical ratio of the intake volume of an air compressor to the least volume at discharge. The least volume is taken as one in the ratio.

Displacement of a Cylinder — the volume swept through by the piston during its stroke. This is expressed in L/s.

Displacement of a Multistage Compressor — the displacement of the first stage only, since the same gas passes through all stages in series.

Double-acting Compressors — a compressor which has two compression strokes per revolution of crankshaft per cylinder, i.e., both faces of the piston are working faces.

Energy — the capacity for doing work; taking a number of forms that may be transformed from one into another, such as thermal (heat), mechanical (work), electrical, and chemical; in customary units, measured in kilowatt-hours (kWh) or megajoules (MJ).

Enthalpy — a measure of the heat energy per unit mass of a material, expressed as kJ/kg.

Free Air — the air at standard atmospheric conditions of 20°C and 101.325 kPa(absolute).

Gauge Pressure — pressure above atmospheric expressed as kPa(gauge).

Higher Heating Value — the gross (Higher Heating Value) energy content of a fuel in MJ. Refer to specific company/supplier data or Appendix C.

Isothermal Compression — compression occurring at constant temperature. This is an ideal condition that does not occur in practical applications.

Kinetic Energy — energy of motion, or the amount of work that can be derived from a moving object by bringing it to rest.

Loaded — when a device or system is operating at maximum or full load capacity.

Low-pressure Orifice Test — a standard method of accurately measuring the air delivered by a compressor.

Maximum Working Pressure(MWP) — the maximum pressure permissible in a device under any circumstances during operation, at a specified temperature. It is a designed safe maximum limit.

Multistage Compressor — a compressor in which compression from initial to final pressure is completed in two or more distinct steps or stages.

Power — the rate at which energy is expended. Expressed in kilowatts.

Power Factor — the ratio of the usable power passing through a circuit to the product of the voltage and current.

Saturation Temperature — the temperature at which a liquid can change to a vapor at a particular pressure.

Single-acting Compressor — a compressor having one compression stroke per revolution of the crank for each cylinder.

Single-stage Compressor — a compressor in which compression from initial to final pressure is complete in a single step or stage.

Specific Heat — the ratio of the amount of heat required to raise the temperature of a given mass of any substance one degree to the quantity required to raise the temperature of an equal mass of a standard substance (usually water at 15°C) one degree.

Specific Volume — the volume of a substance per unit mass; the reciprocal of density.

Specific Weight — weight per unit volume (sometimes called density).

Standard Air Density — 1.204 kg/m^3 at sea level (101.325 kPa(absolute) barometric pressure), dry air, and 20°C.

Steam Rate — the amount of steam required by a turbine to produce a given unit output.

Thermal Efficiency — the ratio of heat energy output to heat energy input.

Turbine Efficiency — the measure of how well a turbine compares with an ideal turbine having no losses.

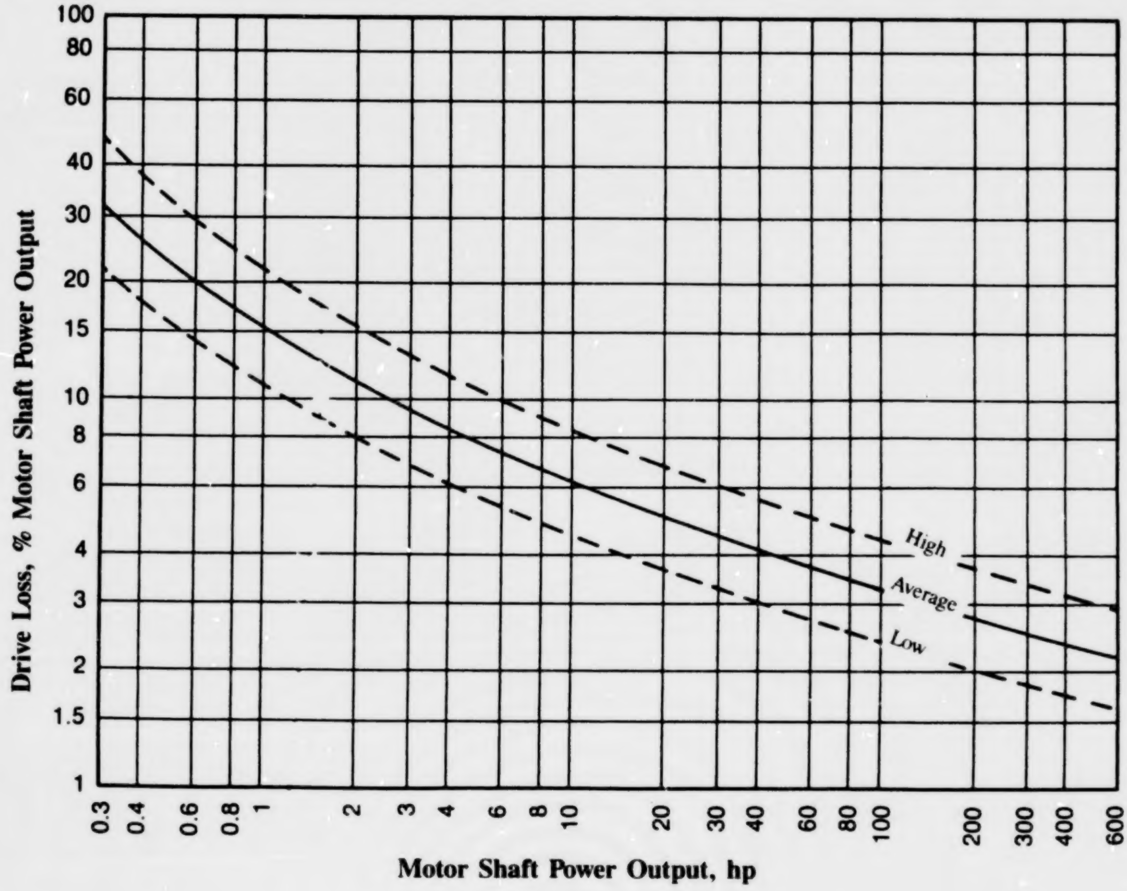
Two-stage Compressor — a compressor in which compression from initial to final pressure is completed in two distinct steps or stages.

Unloaded — when a device or system is operating under no load conditions.

Vacuum Pump — a machine for removing air or gas from a container or piping system for the purpose of maintaining the pressure in the system below atmospheric pressure.

Volumetric Efficiency — the ratio of the actual volume of gas moved by the compressor to the actual piston displacement of the compressor.

ESTIMATED DRIVE LOSS
TABLE 1



Conversion: hp = $\frac{kW}{0.7457}$

TYPICAL PORTION OF INPUT ENERGY RELEASED BY COMPRESSOR COMPONENTS
TABLE 2

Component	Heat Exchange Efficiencies (decimal) (portion of energy available for recovery) Cooling Media: Air	Water
Electric Motor	0.08	—
Intercooler	0.45	0.44
Aftercooler	0.45	0.44
Air cooled compressor	0.05	—
Water cooled compressor	0.01	0.04
Air cooled compressor & water cooled intercooler	0.03	0.44
Air cooled compressor & intercooler	0.50	—

COMMON CHARACTERISTICS OF RECIPROCATING COMPRESSORS
TABLE 3

Compressor	Maximum Capacity (L/s)	Maximum Pressure [kPa(gauge)]	Maximum Power (kW)
Trunk Type			
single stage	20	1040	11
multistage	320	1724	93
Sliding Crosshead			
V type	755	860	120
L & horizontal	4700	3500	3700
Diaphragm	2	420	1

COMMON CHARACTERISTICS OF ROTARY COMPRESSORS
TABLE 4

Compressor	Maximum Capacity (L/s)	Maximum Pressure [kPa(gauge)]	Maximum Power (kW)
Rotary Screw			
small	78	860	30
large	9 400	1030	600
Lobe	10 856	100	697
Vane			
single-stage	850	310	190
double-stage	2 800	1030	300
Liquid ring	4 720	103	746

COMMON CHARACTERISTICS OF DYNAMIC COMPRESSORS
TABLE 5

Compressor	Maximum Capacity (L/s)	Maximum Pressure [kPa(gauge)]	Maximum Power (kW)
Centrifugal	71 000	1 034	10 000
Axial	80 000	1 034	11 000

PROPERTIES OF SATURATED STEAM AND SATURATED WATER (TEMPERATURE)
TABLE 6

Temperature		Press.	Volume, m^3/kg			Enthalpy, kJ/kg			Entropy, kJ/kg K		
°C	K	kPa	Water	Evap.	Steam	Water	Evap.	Steam	Water	Evap.	Steam
t	T	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0.	273.15	0.6108	0.0010002	206.30	206.31	-0.04	2501.6	2501.6	-0.0002	9.1579	9.1577
0.01	273.16	0.6112	0.0010002	206.16	206.16	0.00	2501.6	2501.6	0.0000	9.1575	9.1575
1.0	274.15	0.6566	0.0010001	192.61	192.61	4.17	2499.2	2503.4	0.0153	9.1158	9.1311
2.0	275.15	0.7055	0.0010001	179.92	179.92	8.39	2496.8	2505.2	0.0306	9.0741	9.1047
3.0	276.15	0.7575	0.0010001	168.17	168.17	12.60	2494.5	2507.1	0.0459	9.0326	9.0785
4.0	277.15	0.8129	0.0010000	157.27	157.27	16.80	2492.1	2508.9	0.0611	8.9915	9.0526
5.0	278.15	0.8718	0.0010000	147.16	147.16	21.01	2489.7	2510.7	0.0762	8.9507	9.0269
6.0	279.15	0.9345	0.0010000	137.78	137.78	25.21	2487.4	2512.6	0.0913	8.9102	9.0015
7.0	280.15	1.0012	0.0010001	129.06	129.06	29.41	2485.0	2514.4	0.1063	8.8699	8.9762
8.0	281.15	1.0720	0.0010001	120.96	120.97	33.60	2482.6	2516.2	0.1213	8.8300	8.9513
9.0	282.15	1.1472	0.0010002	113.43	113.44	37.80	2480.3	2518.1	0.1362	8.7903	8.9265
10.0	283.15	1.2270	0.0010003	106.43	106.43	41.99	2477.9	2519.9	0.1510	8.7510	8.9020
12.0	285.15	1.4014	0.0010004	93.83	93.84	50.38	2473.2	2523.6	0.1805	8.6731	8.8536
14.0	287.15	1.5973	0.0010007	82.90	82.90	58.75	2468.5	2527.2	0.2098	8.5963	8.8060
16.0	289.15	1.8168	0.0010010	73.38	73.38	67.13	2463.8	2530.9	0.2388	8.5205	8.7593
18.0	291.15	2.0624	0.0010013	65.09	65.09	75.50	2459.0	2534.5	0.2677	8.4458	8.7135
20.0	293.15	2.337	0.0010017	57.84	57.84	83.86	2454.3	2538.2	0.2963	8.3721	8.6684
22.0	295.15	2.642	0.0010022	51.49	51.49	92.23	2449.6	2541.8	0.3247	8.2994	8.6241
24.0	297.15	2.982	0.0010026	45.92	45.93	100.59	2444.9	2545.5	0.3530	8.2277	8.5806
26.0	299.15	3.360	0.0010032	41.03	41.03	108.95	2440.2	2549.1	0.3810	8.1569	8.5379
28.0	301.15	3.778	0.0010037	36.73	36.73	117.31	2435.4	2552.7	0.4088	8.0870	8.4959
30.0	303.15	4.241	0.0010043	32.93	32.93	125.66	2430.7	2556.4	0.4365	8.0181	8.4546
32.0	305.15	4.753	0.0010049	29.57	29.57	134.02	2425.9	2560.0	0.4640	7.9500	8.4140
34.0	307.15	5.318	0.0010056	26.60	26.60	142.38	2421.2	2563.6	0.4913	7.8828	8.3740
36.0	309.15	5.940	0.0010063	23.97	23.97	150.74	2416.4	2567.2	0.5184	7.8164	8.3348
38.0	311.15	6.624	0.0010070	21.63	21.63	159.09	2411.7	2570.8	0.5453	7.7509	8.2962
40.0	313.15	7.375	0.0010078	19.545	19.546	167.45	2406.9	2574.4	0.5721	7.6861	8.2583
42.0	315.15	8.198	0.0010086	17.691	17.692	175.81	2402.1	2577.9	0.5987	7.6222	8.2209
44.0	317.15	9.100	0.0010094	16.035	16.036	184.17	2397.3	2581.5	0.6252	7.5590	8.1842
46.0	319.15	10.086	0.0010103	14.556	14.557	192.53	2392.5	2585.1	0.6514	7.4966	8.1481
48.0	321.15	11.162	0.0010112	13.232	13.233	200.89	2387.7	2588.6	0.6776	7.4350	8.1125
50.0	323.15	12.335	0.0010121	12.045	12.046	209.26	2382.9	2592.2	0.7035	7.3741	8.0776
52.0	325.15	13.613	0.0010131	10.979	10.980	217.62	2378.1	2595.7	0.7293	7.3138	8.0432
54.0	327.15	15.002	0.0010140	10.021	10.022	225.99	2373.2	2599.2	0.7550	7.2543	8.0093
56.0	329.15	16.511	0.0010150	9.158	9.159	234.35	2368.4	2602.7	0.7804	7.1955	7.9759
58.0	331.15	18.147	0.0010161	8.380	8.381	242.72	2363.5	2606.2	0.8058	7.1373	7.9431
60.0	333.15	19.920	0.0010171	7.678	7.679	251.09	2358.6	2609.7	0.8310	7.0798	7.9108
62.0	335.15	21.838	0.0010182	7.043	7.044	259.46	2353.7	2613.2	0.8560	7.0230	7.8790
64.0	337.15	23.912	0.0010193	6.468	6.469	267.84	2348.8	2616.6	0.8809	6.9667	7.8477
66.0	339.15	26.150	0.0010205	5.947	5.948	276.21	2343.9	2620.1	0.9057	6.9111	7.8168
68.0	341.15	28.563	0.0010217	5.475	5.476	284.59	2338.9	2623.5	0.9303	6.8561	7.7864
70.0	343.15	31.16	0.0010228	5.045	5.046	292.97	2334.0	2626.9	0.9548	6.8017	7.7565
72.0	345.15	33.96	0.0010241	4.655	4.656	301.36	2329.0	2630.3	0.9792	6.7478	7.7270
74.0	347.15	36.96	0.0010253	4.299	4.300	309.74	2324.0	2633.7	1.0034	6.6945	7.6979
76.0	349.15	40.19	0.0010266	3.975	3.976	318.13	2318.9	2637.1	1.0275	6.6418	7.6693
78.0	351.15	43.65	0.0010279	3.679	3.680	326.52	2313.9	2640.4	1.0514	6.5896	7.6410
80.0	353.15	47.36	0.0010292	3.408	3.409	334.92	2308.8	2643.8	1.0753	6.5380	7.6132
82.0	355.15	51.33	0.0010305	3.161	3.162	343.31	2303.8	2647.1	1.0990	6.4868	7.5858
84.0	357.15	55.57	0.0010319	2.934	2.935	351.71	2298.6	2650.4	1.1225	6.4362	7.5588
86.0	359.15	60.11	0.0010333	2.726	2.727	360.12	2293.5	2653.6	1.1460	6.3861	7.5321
88.0	361.15	64.95	0.0010347	2.535	2.536	368.53	2288.4	2656.9	1.1693	6.3365	7.5058
90.0	363.15	70.11	0.0010361	2.3603	2.3613	376.94	2283.2	2660.1	1.1925	6.2873	7.4799
92.0	365.15	75.61	0.0010376	2.1992	2.2002	385.36	2278.0	2663.4	1.2156	6.2387	7.4543
94.0	367.15	81.46	0.0010391	2.0509	2.0519	393.78	2272.8	2666.6	1.2386	6.1905	7.4291
96.0	369.15	87.69	0.0010406	1.9143	1.9153	402.20	2267.5	2669.7	1.2615	6.1427	7.4042
98.0	371.15	94.30	0.0010421	1.7883	1.7893	410.63	2262.2	2672.9	1.2842	6.0954	7.3796
100.0	373.15	101.33	0.0010437	1.6720	1.6730	419.06	2256.9	2676.0	1.3069	6.0485	7.3554

PROPERTIES OF SATURATED STEAM AND SATURATED WATER (TEMPERATURE)

TABLE 6

Temperature		Press.	Volume, m^3/kg			Enthalpy, kJ/kg			Entropy, kJ/kg K		
°C	K	kPa	Water	Evap.	Steam	Water	Evap.	Steam	Water	Evap.	Steam
t	T	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
100.0	373.15	101.33	0.0010437	1.6720	1.6730	419.06	2256.9	2676.0	1.3069	6.0485	7.3554
105.0	378.15	120.80	0.0010477	1.4182	1.4193	440.17	2243.6	2683.7	1.3630	5.9331	7.2962
110.0	383.15	143.27	0.0010519	1.2089	1.2099	461.32	2230.0	2691.3	1.4185	5.8203	7.2388
115.0	388.15	169.06	0.0010562	1.0352	1.0363	482.50	2216.2	2698.7	1.4733	5.7099	7.1832
120.0	393.15	198.54	0.0010606	0.8905	0.8915	503.72	2202.2	2706.0	1.5276	5.6017	7.1293
125.0	398.15	232.1	0.0010652	0.7692	0.7702	524.99	2188.0	2713.0	1.5813	5.4957	7.0769
130.0	403.15	270.1	0.0010700	0.6671	0.6681	546.31	2173.6	2719.9	1.6344	5.3917	7.0261
135.0	408.15	313.1	0.0010750	0.5807	0.5818	567.68	2158.9	2726.6	1.6869	5.2897	6.9766
140.0	413.15	361.4	0.0010801	0.5074	0.5085	589.10	2144.0	2733.1	1.7390	5.1894	6.9284
145.0	418.15	415.5	0.0010853	0.4449	0.4460	610.59	2128.7	2739.3	1.7906	5.0910	6.8815
150.0	423.15	476.0	0.0010908	0.3914	0.3924	632.15	2113.2	2745.4	1.8416	4.9941	6.8358
155.0	428.15	543.3	0.0010964	0.3453	0.3464	653.77	2097.4	2751.2	1.8923	4.8989	6.7911
160.0	433.15	618.1	0.0011022	0.3057	0.3068	675.47	2081.3	2756.7	1.9425	4.8050	6.7475
165.0	438.15	700.8	0.0011082	0.2713	0.2724	697.25	2064.8	2762.0	1.9923	4.7126	6.7048
170.0	443.15	792.0	0.0011145	0.2414	0.2426	719.12	2047.9	2767.1	2.0416	4.6214	6.6630
175.0	448.15	892.4	0.0011209	0.21542	0.21654	741.07	2030.7	2771.8	2.0906	4.5314	6.6221
180.0	453.15	1002.7	0.0011275	0.19267	0.19380	763.12	2013.2	2776.3	2.1393	4.4426	6.5819
185.0	458.15	1123.3	0.0011344	0.17272	0.17386	785.26	1995.2	2780.4	2.1876	4.3548	6.5424
190.0	463.15	1255.1	0.0011415	0.15517	0.15632	807.52	1976.7	2784.3	2.2356	4.2680	6.5036
195.0	468.15	1398.7	0.0011489	0.13969	0.14084	829.88	1957.9	2787.8	2.2833	4.1821	6.4654
200.0	473.15	1554.9	0.0011565	0.12600	0.12716	852.37	1938.6	2790.9	2.3307	4.0971	6.4278
205.0	478.15	1724.3	0.0011644	0.11386	0.11503	874.99	1918.8	2793.8	2.3778	4.0128	6.3906
210.0	483.15	1907.7	0.0011726	0.10307	0.10424	897.73	1898.5	2796.2	2.4247	3.9293	6.3539
215.0	488.15	2106.0	0.0011811	0.09344	0.09463	920.63	1877.6	2798.3	2.4713	3.8463	6.3176
220.0	493.15	2319.8	0.0011900	0.08485	0.08604	943.67	1856.2	2799.9	2.5178	3.7639	6.2817
225.0	498.15	2550.	0.0011992	0.07715	0.07835	966.88	1834.3	2801.2	2.5641	3.6820	6.2461
230.0	503.15	2798.	0.0012087	0.07024	0.07145	990.27	1811.7	2802.0	2.6102	3.6006	6.2107
235.0	508.15	3063.	0.0012187	0.06403	0.06525	1013.83	1788.5	2802.3	2.6561	3.5194	6.1756
240.0	513.15	3348.	0.0012291	0.05843	0.05965	1037.60	1764.6	2802.2	2.7020	3.4386	6.1406
245.0	518.15	3652.	0.0012399	0.05337	0.05461	1061.58	1740.0	2801.6	2.7478	3.3579	6.1057
250.0	523.15	3978.	0.0012513	0.04879	0.05004	1085.78	1714.7	2800.4	2.7935	3.2773	6.0708
255.0	528.15	4325.	0.0012632	0.04463	0.04590	1110.23	1688.5	2798.7	2.8392	3.1968	6.0359
260.0	533.15	4694.	0.0012756	0.04086	0.04213	1134.94	1661.5	2796.4	2.8848	3.1161	6.0010
265.0	538.15	5088.	0.0012887	0.03742	0.03871	1159.93	1633.5	2793.5	2.9306	3.0353	5.9658
270.0	543.15	5506.	0.0013025	0.03429	0.03559	1185.23	1604.6	2789.9	2.9763	2.9541	5.9304
275.0	548.15	5950.	0.0013170	0.03142	0.03274	1210.86	1574.7	2785.5	3.0222	2.8725	5.8947
280.0	553.15	6420.	0.0013324	0.02879	0.03013	1236.84	1543.6	2780.4	3.0683	2.7903	5.8586
285.0	558.15	6919.	0.0013487	0.02638	0.02773	1263.21	1511.3	2774.5	3.1146	2.7074	5.8220
290.0	563.15	7446.	0.0013659	0.02417	0.02554	1290.01	1477.6	2767.6	3.1611	2.6237	5.7848
295.0	568.15	8004.	0.0013844	0.02213	0.02351	1317.27	1442.6	2759.8	3.2079	2.5389	5.7469
300.0	573.15	8593.	0.0014041	0.020245	0.021649	1345.05	1406.0	2751.0	3.2552	2.4529	5.7081
305.0	578.15	9214.	0.0014252	0.018502	0.019927	1373.40	1367.7	2741.1	3.3029	2.3656	5.6685
310.0	583.15	9870.	0.0014480	0.016886	0.018334	1402.39	1327.6	2730.0	3.3512	2.2766	5.6278
315.0	588.15	10561.	0.0014726	0.015383	0.016856	1432.09	1285.5	2717.6	3.4002	2.1856	5.5858
320.0	593.15	11289.	0.0014995	0.013980	0.015480	1462.60	1241.1	2703.7	3.4500	2.0923	5.5423
325.0	598.15	12056.	0.0015289	0.012666	0.014195	1494.93	1194.0	2688.0	3.5008	1.9961	5.4969
330.0	603.15	12863.	0.0015615	0.011428	0.012989	1526.52	1143.6	2670.2	3.5528	1.8962	5.4490
335.0	608.15	13712.	0.0015978	0.010256	0.011854	1560.25	1089.5	2649.7	3.6063	1.7916	5.3979
340.0	613.15	14605.	0.0016387	0.009142	0.010780	1595.47	1030.7	2626.2	3.6616	1.6811	5.3427
345.0	618.15	15545.	0.0016858	0.008077	0.009763	1632.52	966.4	2598.9	3.7193	1.5636	5.2828
350.0	623.15	16535.	0.0017411	0.007058	0.008799	1671.94	895.7	2567.7	3.7800	1.4376	5.2177
355.0	628.15	17577.	0.0018085	0.006051	0.007859	1716.63	813.8	2530.4	3.8489	1.2953	5.1442
360.0	633.15	18675.	0.0018959	0.005044	0.006940	1764.17	721.3	2485.4	3.9210	1.1390	5.0600
365.0	638.15	19833.	0.0020160	0.003996	0.006012	1817.96	610.0	2428.0	4.0021	0.9558	4.9579
370.0	643.15	21054.	0.0022136	0.002759	0.004973	1890.21	452.6	2342.8	4.1108	0.7036	4.8144
371.0	644.15	21306.	0.0022778	0.002446	0.004723	1910.50	407.4	2317.9	4.1414	0.6324	4.7738
372.0	645.15	21562.	0.0023636	0.002075	0.004439	1935.57	351.4	2287.0	4.1794	0.5446	4.7240
373.0	646.15	21820.	0.0024963	0.001588	0.004084	1970.50	273.5	2244.0	4.2326	0.4233	4.6559
374.0	647.15	22081.	0.0028427	0.000623	0.003466	2046.72	109.5	2156.2	4.3493	0.1692	4.5185
374.15	647.30	22120.	0.00317	0.0	0.00317	2107.37	0.0	2107.4	4.4429	0.0	4.4429

EFFECTS ON TURBINE STEAM RATE
TABLE 7

Effect on Steam Rate of Inlet Pressure and Superheat[a]

Inlet Pressure		Inlet Superheat		Exhaust Pressure		Design Output		Design Speed		Design Steam Rate		Steam Rate
psig	kPa gauge	F	°C	in. Hg vac	kPa abs	hp	kW	rpm	r/s	lb/hp-h	kg/ kWh	%
125	850	0	0	26	13.5	500	375	6000	100	16.5	10.1	100
125	850	100	55	26	13.5	500	375	6000	100	15.5	9.4	93
125	850	200	110	26	13.5	500	375	6000	100	14.6	8.9	88
250	1725	0	0	26	13.5	500	375	6000	100	14.5	8.8	87
250	1725	100	55	26	13.5	500	375	6000	100	13.6	8.3	82
250	1725	200	110	26	13.5	500	375	6000	100	12.7	7.7	77
400	2750	0	00	26	13.5	500	375	6000	100	13.6	8.2	82
400	2750	100	55	26	13.5	500	375	6000	100	12.5	7.6	75
400	2750	200	110	26	13.5	500	375	6000	100	11.7	7.1	70

[a] Number of stages and efficiency are constant.

Effect on Steam Rate of Power and Speed[a]

Inlet Pressure		Inlet Superheat		Exhaust Pressure		Design Output		Design Speed		Design Steam Rate		Steam Rate
psig	kPa gauge	F	°C	in. Hg vac	kPa abs	hp	kW	rpm	r/s	lb/hp-h	kg/ kWh	%
250	1725	0	0	26	13.5	500	375	3600	60	18.5	11.2	100
250	1725	0	0	26	13.5	500	375	6000	100	14.5	8.8	78
250	1725	0	0	26	13.5	1500	1125	3600	60	16.0	9.7	86
250	1725	0	0	26	13.5	1500	1125	5000	83	13.5	8.2	73
250	1725	0	0	26	13.5	3500	2625	4000	67	12.5	7.6	68
250	1725	0	0	26	13.5	3500	2625	6000	100	11.5	7.0	62

[a] Number of stages is constant.

Effect on Steam Rate of Exhaust Pressure[a]

Inlet Pressure		Inlet Superheat		Exhaust Pressure		Design Output		Design Speed		Design Steam Rate		Steam Rate
psig	kPa gauge	F	°C		kPa abs	hp	kW	rpm	r/s	lb/hp-h	kg/ kWh	%
250	1725	0	0	26 in. Hg vac	13.5	1000	750	5000	83	14.0	8.5	100
250	1725	0	0	28 in. Hg vac	6.8	1000	750	5000	83	12.7	7.7	91
250	1725	0	0	5 psig	135	1000	750	5000	83	24.8	15.1	177
250	1725	0	0	20 psig	240	1000	750	5000	83	32.4	19.7	231
250	1725	0	0	50 psig	450	1000	750	5000	83	41.2	25.1	294

[a] Number of stages is constant.

Effect on Steam Rate of Number of Stages

Inlet Pressure		Inlet Superheat		Exhaust Pressure		Design Power		Design Speed		Number of Stages	Design Steam Rate		Steam Rate
psig	kPa gauge	F	°C		kPa abs	hp	kW	rpm	r/s		lb/hp-h	kg/kWh	%
125	850	0	0	26 in. vac	13.5	1000	750	5000	83	5	16.0	9.7	100
125	850	0	0	26 in. vac	13.5	1000	750	5000	83	7	15.1	9.2	94
125	850	0	0	26 in. vac	13.5	1000	750	5000	83	9	14.4	8.8	90
150	1025	0	0	5 psig	135	500	375	6000	100	2	36.0	21.9	225
150	1025	0	0	5 psig	135	500	375	6000	100	3	31.0	18.8	194
150	1025	0	0	5 psig	135	500	375	6000	100	5	29.5	17.9	184

Courtesy of ASHRAE

TYPICAL CAPACITY CHARACTERISTICS OF STEAM TURBINES
TABLE 8

Turbine	Maximum Power (kW)	Maximum Inlet Pressure [kPa(gauge)]	Maximum Outlet Pressure [kPa(gauge)]	Maximum Speed (rpm)
Reheat	4 849	4 134	4 134	8 000
Straight Flow	1 119	9 990	2 067	12 500
Extraction	60 000	10 335	2 756	12 500
Induction	60 000	10 335	2 754	12 500

COMMON CONVERSIONS

1 barrel (35 Imp gal) (42 US gal) = 159.1 litres

1 gallon (Imp) = 1.20094 gallon (US)

1 horsepower (boiler) = 9809.6 watts

1 horsepower = 2545 Btu/hour

1 horsepower = 0.746 kilowatts

1 joule = 1 N-m

Kelvin = (°C + 273.15)

1 kilowatt-hour = 3600 kilojoules

1 Newton = 1 $kg\text{-}m/s^2$

1 therm = 10^5 Btu

1 ton (refrigerant) = 12002.84 Btu/hour

1 ton (refrigerant) = 3516.8 watts

1 watt = 1 joule/second

Rankine = (°F + 459.67)

Cubes

1 yd^3 = 27 ft^3

1 ft^3 = 1728 in^3

1 cm^3 = 1000 mm^3

1 m^3 = 10^6 cm^3

1 m^3 = 1000 L

Squares

1 yd^2 = 9 ft^2

1 ft^2 = 144 in^2

1 cm^2 = 100 mm^2

1 m^2 = 10000 cm^2

SI PREFIXES

Prefix	Symbol	Magnitude	Factor
tera	T	1 000 000 000 000	10^{12}
giga	G	1 000 000 000	10^9
mega	M	1 000 000	10^6
kilo	k	1 000	10^3
hecto	h	100	10^2
deca	da	10	10^1
deci	d	0.1	10^{-1}
centi	c	0.01	10^{-2}
milli	m	0.001	10^{-3}
micro	u	0.000 001	10^{-6}
nano	n	0.000 000 001	10^{-9}
pica	p	0.000 000 000 001	10^{-12}

UNIT CONVERSION TABLES
METRIC TO IMPERIAL

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
amperes/square centimetre	A/cm^2	amperes/square inch	A/in^2	6.452
Celsius	°C	Fahrenheit	°F	(°C × 9/5) + 32
centimetres	cm	inches	in	0.3937
cubic centimetres	cm^3	cubic inches	in^3	0.06102
cubic metres	m^3	cubic foot	ft^3	35.314
grams	g	ounces	oz	0.03527
grams	g	pounds	lb	0.0022
grams/litre	g/L	pounds/cubic foot	lb/ft^3	0.06243
joules	J	Btu	Btu	9.480×10^{-4}
joules	J	foot-pounds	ft-lb	0.7376
joules	J	horsepower-hours	hp-h	3.73×10^{-7}
joules/metre, (Newtons)	J/m, N	pounds	lb	0.2248
kilograms	kg	pounds	lb	2.205
kilograms	kg	tons (long)	ton	9.842×10^{-4}
kilograms	kg	tons (short)	tn	1.102×10^{-3}
kilometres	km	miles (statute)	mi	0.6214
kilopascals	kPa	atmospheres	atm	9.87×10^{-3}
kilopascals	kPa	inches of mercury (@ 32°F)	in Hg	0.2953
kilopascals	kPa	inches of water (@ 4°C)	in H_2O	4.0147
kilopascals	kPa	pounds/square inch	psi	0.1450
kilowatts	kW	foot-pounds/second	ft-lb/s	737.6
kilowatts	kW	horsepower	hp	1.341
kilowatt-hours	kWh	Btu	Btu	3413
litres	L	cubic foot	ft^3	0.03531
litres	L	gallons (Imp)	gal (Imp)	0.21998
litres	L	gallons (US)	gal (US)	0.2642
litres/second	L/s	cubic foot/minute	cfm	2.1186
lumen/square metre	lm/m^2	lumen/square foot	lm/ft^2	0.09290
lux, lumen/square metre	lx, lm/m^2	footcandles	fc	0.09290
metres	m	foot	ft	3.281
metres	m	yard	yd	1.09361
parts per million	ppm	grains/gallon (Imp)	gr/gal (Imp)	0.07
parts per million	ppm	grains/gallon (US)	gr/gal (US)	0.05842
permeance (metric)	PERM	permeance (Imp)	perm	0.01748
square centimetres	cm^2	square inches	in^2	0.1550
square metres	m^2	square foot	ft^2	10.764
square metres	m^2	square yards	yd^2	1.196
tonne (metric)	t	pounds	lb	2204.6
watt	W	Btu/hour	Btu/h	3.413
watt	W	lumen	lm	668.45

4138T/T33(-4)

UNIT CONVERSION TABLES
IMPERIAL TO METRIC

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
ampere/in^2	A/in^2	ampere/cm^2	A/cm^2	0.1550
atmospheres	atm	kilopascals	kPa	101.325
British Thermal Unit	Btu	joules	J	1054.8
Btu	Btu	kilogram-metre	kg-m	107.56
Btu	Btu	kilowatt-hour	kWh	2.928×10^{-4}
Btu/hour	Btu/h	watt	W	0.2931
calorie, gram	cal or g-cal	joules	J	4.186
chain	chain	metre	m	20.11684
cubic foot	ft^3	cubic metre	m^3	0.02832
cubic foot	ft^3	litre	L	28.32
cubic foot/minute	cfm	litre/second	L/s	0.47195
cycle/second	c/s	Hertz	Hz	1.00
Fahrenheit	°F	Celsius	°C	(°F-32)/1.8
foot	ft	metre	m	0.3048
footcandle	fc	lux, lumen/ square metre	lx, lm/m^2	10.764
footlambert	fL	candela/square metre	cd/m^2	3.42626
foot-pounds	ft-lb	joule	J	1.356
foot-pounds	ft-lb	kilogram-metres	kg-m	0.1383
foot-pounds/second	ft-lb/s	kilowatt	kW	1.356×10^{-3}
gallons (Imp)	gal (Imp)	litres	L	4.546
gallons (US)	gal (US)	litres	L	3.785
grains/gallon (Imp)	gr/gal (Imp)	parts per million	ppm	14.286
grains/gallon (US)	gr/gal (US)	parts per million	ppm	17.118
horsepower	hp	watts	W	745.7
horsepower-hours	hp-h	joules	J	2.684×10^6
inches	in	centimetres	cm	2.540
inches of Mercury (@ 32°F)	in Hg	kilopascals	kPa	3.386
inches of water (@ 4°C)	in H_2O	kilopascals	kPa	0.2491

4138T/T33(-1)

UNIT CONVERSION TABLES
IMPERIAL TO METRIC (cont'd)

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
lamberts	*L	candela/square metre	cd/m^2	3.183
lumen/square foot	lm/ft^2	lumen/square metre	lm/m^2	10.76
lumen	lm	watt	W	0.001496
miles (statute)	mi	kilometres	km	1.6093
ounces	oz	grams	g	28.35
perm (at 0°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.721×10^{-11}
perm (at 23°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.745×10^{-11}
perm-inch (at 0°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4532×10^{-12}
perm-inch (at 23°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4593×10^{-12}
pint (Imp)	pt	litre	L	0.56826
pounds	lb	grams	g	453.5924
pounds	lb	joules/metre, (Newtons)	J/m, N	4.448
pounds	lb	kilograms	kg	0.4536
pounds	lb	tonne (metric)	t	4.536×10^{-4}
pounds/cubic foot	lb/ft^3	grams/litre	g/L	16.02
pounds/square inch	psi	kilopascals	kPa	6.89476
quarts	qt	litres	L	1.1365
slug	slug	kilograms	kg	14.5939
square foot	ft^2	square metre	m^2	0.09290
square inches	in^2	square centimetres	cm^2	6.452
square yards	yd^2	square metres	m^2	0.83613
tons (long)	ton	kilograms	kg	1016
tons (short)	tn	kilograms	kg	907.185
yards	yd	metres	m	0.9144

* "L" as used in Lighting

The following typical values for conversion factors may be used when actual data are unavailable. The MJ and Btu equivalencies are heats of combustion. Hydrocarbons are shown at the higher heating value, wet basis. Some items listed are typically feedstocks, but are included for completeness and as a reference source. The conversion factors for coal are approximate since the heating value of a specific coal is dependent on the particular mine from which it is obtained.

ENERGY TYPE	METRIC	IMPERIAL
COAL		
— metallurgical	29,000 megajoules/tonne	25.0 × 10^6 Btu/ton
— anthracite	30,000 megajoules/tonne	25.8 × 10^6 Btu/ton
— bituminous	32,100 megajoules/tonne	27.6 × 10^6 Btu/ton
— sub-bituminous	22,100 megajoules/tonne	19.0 × 10^6 Btu/ton
— lignite	16,700 megajoules/tonne	14.4 × 10^6 Btu/ton
COKE		
— metallurgical	30,200 megajoules/tonne	26.0 × 10^6 Btu/ton
— petroleum		
— raw	23,300 megajoules/tonne	20.0 × 10^6 Btu/ton
— calcined	32,600 megajoules/tonne	28.0 × 10^6 Btu/ton
PITCH	37,200 megajoules/tonne	32.0 × 10^6 Btu/ton
CRUDE OIL	38,5 megajoules/litre	5.8 × 10^6 Btu/bbl
No. 2 OIL	38.68 megajoules/litre	5.88 × 10^6 Btu/bbl
		.168 × 10^6 Btu/IG
No. 4 OIL	40.1 megajoules/litre	6.04 × 10^6 Btu/bbl
		.173 × 10^6 Btu/IG
No. 6 OIL (RESID. BUNKER C)		
@ 2.5% sulphur	42.3 megajoules/litre	6.38 × 10^6 Btu/bbl
		.182 × 10^6 Btu/IG
@ 1.0% sulphur	40.5 megajoules/litre	6.11 × 10^6 Btu/bbl
		.174 × 10^6 Btu/IG
@ .5% sulphur	40.2 megajoules/litre	6.05 × 10^6 Btu/bbl
		.173 × 10^6 Btu/IG
KEROSENE	37.68 megajoules/litre	.167 × 10^6 Btu/IG
DIESEL FUEL	38.68 megajoules/litre	.172 × 10^6 Btu/IG
GASOLINE	36.2 megajoules/litre	.156 × 10^6 Btu/IG
NATURAL GAS	37.2 megajoules/m^3	1.00 × 10^6 Btu/MCF
PROPANE	50.3 megajoules/kg	.02165 × 10^6 Btu/lb
	26.6 megajoules/litre	.1145 × 10^6 Btu/IG
ELECTRICITY	3.6 megajoules/kWh	.003413 × 10^6 Btu/kWh

Compressor Inlet Air Temperature

Worksheet 14-C1

(Page 1 of 1)

Company: ____________________ Date: ____________________

Location: ____________________ By: ____________________

Initial inlet temperature, T_{i1} ____________ K

Revised inlet temperature, T_{i2} ____________ K

Initial motor power input, Wm_{i1} ____________ /kW

Unit electrical energy cost, Ce $ ____________ /kWh

Operation time, h ____________ h/yr

Revised motor power input,

$$Wm_{i2} = Wm_{i1} \times (1 + [0.00341 \times (T_{i2} - T_{i1})])$$

$$= \quad \times (1 + [0.00341 \times (\quad - \quad)])$$

$$= \underline{\qquad\qquad} \text{ kW}$$

$$\text{Annual energy cost saving} = (Wm_{i1} - Wm_{i2}) \times h \times Ce$$

$$= (\quad - \quad) \times \quad \times$$

$$= \$ \underline{\qquad\qquad\qquad} \text{/yr}$$

Compressor Inlet Pressure

Worksheet 14-C2
(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data:

Number of stages, N ____________

Discharge pressure, P_d ____________ kPa(absolute)

Initial inlet pressure, P_{i1} ____________ kPa(absolute)

Revised inlet pressure, P_{i2} ____________ kPa(absolute)

Initial motor power input, Wm_{i1} ____________ kW

Unit electrical energy cost, Ce $ ____________ /kWh

Operation time, h ____________ h/yr

Factor of initial inlet pressure,

$$F_{i1} = \left(\frac{P_d}{P_{i1}}\right)^{\frac{0.231}{N}} - 1$$

$$= \left(\frac{\quad}{\quad}\right)^{\frac{0.231}{\quad}} - 1$$

$$= \underline{\qquad\qquad}$$

Compressor Inlet Pressure

Worksheet 14-C2

Page 2 of 2

Company: ______________________________ Date: ______________________________

Location: ______________________________ By: ______________________________

Factor of revised inlet pressure,

$$F_{i2} = \left(\frac{P_d}{P_{i2}}\right)^{\frac{0.231}{N}} - 1$$

$$= \left(\frac{}{}\right)^{\frac{0.231}{}} - 1$$

= ______________

Revised motor power input, $Wm_{i2} = Wm_{i1} \times \frac{P_{i1} \times F_{i1}}{P_{i2} \times F_{i2}}$

$$= \quad \times \frac{\quad \times \quad}{\quad \times \quad}$$

= ______________ kW

Annual energy cost saving = $(Wm_{i1} - Wm_{i2}) \times h \times Ce$

= (−   ) x   x

= $______________/yr

Cooling Water Saving

Worksheet 14-C3
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Cooling water flow rate, f_w ______________ L/s

Unit water cost, Cw $ ______________ /m^3

Operation time, h ______________ h/yr

Annual water cost saving = f_w x Cw x h x 3.6

= x x x 3.6

= $______________/yr

Compressor Drive
Worksheet 14-C4
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Motor shaft power output, Wm_o ______________ kW

Turbine shaft power output, Wt_o ______________ kW

Drive loss (Table 1) ______________ %

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$= 1 - \frac{\_\_\_\_\_\_}{100}$

= ______________

Power input to the compressor shaft, $Wc_i = Wm_o \times Ef_d$

or, $Wc_i = Wt_o \times Ef_d$

= x

= ______________ kW

Compressor Component Heating/Cooling

Worksheet 14-C5

(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Power input to the compressor shaft, Wc_i ____________ kW

Cooling media ____________

Type of fuel ____________

Fuel higher heating value, HHV (Appendix C) ____________ MJ/ ____

Unit fuel, steam or electrical energy cost, C____ $ ____________ / ______

Operation time, h ____________ h/yr

Efficiency of fuel burning device, Ef_b (if unknown, use 0.75) ____________ (decimal)

Portion of input energy released

Component	Heat Exchange Efficiency Table 2
____________	____________
____________	____________
____________	____________
____________	____________
____________	____________
Total, Ef_e	____________

Potential heat recovery, $Q_{pr} = 3.6 \times Wc_i \times Ef_e$

$= 3.6 \times \quad \times$

$=$ ____________ MJ/h

Compressor Component Heating/Cooling

Worksheet 14-C5

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

$$\text{Annual fuel cost saving} = \frac{Q_{pr} \times Cf \times h}{HHV \times Ef_b}$$

$$= \frac{\quad \times \quad \times \quad}{\quad \times \quad}$$

= $______________/yr

$$\text{Annual steam cost saving} = \frac{Q_{pr} \times Cs \times h}{2.2}$$

$$= \frac{\quad \times \quad \times \quad}{2.2}$$

= $______________/yr

Annual cooling energy cost saving (if Sc is unknown use 80 kWh/GJ for an electrical cooling system and 0.45 kg/GJ for a system operated by steam)

$$\text{Annual Savings} = \frac{Q_{pr} \times C_{\_} \times Sc \times h}{1000}$$

$$= \frac{\quad \times \quad \times \quad \times \quad}{1000}$$

= $______________/yr

Equivalent Free Air Flow

Worksheet 14-C6
Page 1 of 1

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Data:

Initial air flow rate, f_{a1} ______________ L/s

Initial temperature, T_1 ______________ K

Initial pressure, P_1 ______________ kPa(absolute)

$$\text{Equivalent free air flow rate, } f_{as} = \frac{P_1 \times f_{a1} \times 293.15}{T_1 \times 101.325}$$

$$= \frac{\underline{\qquad} \times \underline{\qquad} \times 293.15}{\underline{\qquad} \times 101.325}$$

= ______________ L/s

Variable Speed Drive

Worksheet 14-C7
(Page 1 of 2)

Company: ____________________ Date: ____________________

Location: ____________________ By: ____________________

Data:

Motor shaft power output-loaded, Wm_{oL}	__________	kW
Rated motor efficiency, Ef_{mr}	__________	(decimal)
Equivalent operation time — fully loaded, h_L	__________	h/yr
Operation time at variable speed, h_v	__________	h/yr
Efficiency of variable speed controller, Ef_{vc}	__________	(decimal)
Compressor drive loss (Table 1)	__________	%
Unit electrical energy cost, Ce	$ __________	/kWh

Compressor drive efficiency, $Ef_d = 1 - \frac{\text{Drive Loss}}{100}$

$= 1 - \frac{\rule{2em}{0.4pt}}{100}$

$=$ __________

Power input to compressor shaft – fully loaded, $Wc_{iL} = Wm_{oL} \times Ef_d$

$= \quad \times$

$=$ __________ kW

Average power input to compressor shaft, $Wc_{iv} = \frac{Wc_{iL} \times h_L}{h_v}$

$= \frac{\quad \times \quad}{\rule{4em}{0.4pt}}$

$=$ __________ kW

Variable Speed Drive

Worksheet 14-C7

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Average motor power input, $Wm_{iv} = \dfrac{Wc_{iv}}{Ef_m \times Ef_{vd} \times Ef_d}$

$= \dfrac{\qquad}{\quad \times \quad \times \quad}$

$=$ ______________ kW

Annual energy cost $= Wm_{iv} \times h_v \times Ce$

$= \quad \times \quad \times$

= $______________/yr

Gas Turbine Inlet and Outlet Conditions

Worksheet 14-T1

(Page 1 of 2)

Company: ____________________ Date: ____________________

Location: ____________________ By: ____________________

Nameplate turbine shaft power output, Wt_{on} ____________ kW

Altitude, AL ____________ m

Initial inlet temperature, T_{i1} ____________ K

Revised inlet temperature, T_{i2} ____________ K

Initial reduction of static pressure below atmospheric at inlet flange, P_{ri1} ____________ Pa

Revised reduction of static pressure below atmospheric at inlet flange, P_{ri2} ____________ Pa

Initial increase of static pressure above atmospheric at outlet flange, P_{o1} ____________ Pa

Revised increase of static pressure above atmospheric at outlet flange, P_{o2} ____________ Pa

Intial fuel flow rate (measured), f_{f1} ____________ L/s or m^3/s

Unit fuel cost, Cf ____________ \$/L or \$/m^3

Operation time, h ____________ h/yr

Initial corrected turbine shaft power output,

$$Wt_{oc1} = Wt_{on} \text{ x} \left[1 - \frac{(T_{i1} - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri1}}{5000} - \frac{P_{o1}}{8333} \right]$$

$$= \underline{\qquad\qquad} \text{ x} \left[1 - \frac{(\underline{\qquad\quad} - 288.15)}{111} - \frac{\underline{\qquad}}{8696} - \frac{\underline{\qquad}}{5000} - \frac{\underline{\qquad}}{8333} \right]$$

= ____________ kW

Gas Turbine Inlet and Outlet Conditions

Worksheet 14-T1

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Revised corrected turbine shaft power output,

$$Wt_{oc2} = Wt_{on} \times \left[1 - \frac{(T_{i2} - 288.15)}{111} - \frac{AL}{8696} - \frac{P_{ri2}}{5000} - \frac{P_{o2}}{8333} \right]$$

$$= \qquad \times \left[1 - \frac{(\qquad - 288.15)}{111} - \frac{\underline{\qquad}}{8696} - \frac{\underline{\qquad}}{5000} - \frac{\underline{\qquad}}{8333} \right]$$

$$= \underline{\qquad\qquad} \text{ kW}$$

Revised fuel flow rate, $f_{f2} = f_{f1} \times \frac{Wt_{oc1}}{Wt_{oc2}}$

$$= \qquad \times \underline{\qquad}$$

$$= \underline{\qquad\qquad} \text{ L/s or m}^3\text{/s}$$

Annual fuel cost saving $= (f_{f1} - f_{f2}) \times h \times Cf \times 3600$

$$= (\qquad - \qquad) \times \qquad \times \qquad \times 3600$$

$$= \$\underline{\qquad\qquad}\text{/yr}$$

Gas Turbine Oil Cooler Heat Recovery

Worksheet 14-T2
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Fuel flow rate (measured), f_f ______________ L/s or m^3/s

Oil heat rejection factor (4.44 for basic cycle or 6.55 for regeneration cycle) ______________

Operation time (that recovered heat usable), h ______________ h/yr

Fuel higher heating value, HHV (Appendix C) ______________ MJ/L or MJ/m^3

Unit steam cost, Cs $ ______________ /kg

Displaced steam heating value ______________ kJ/kg (1)

Energy input to gas turbine, $Q_g = f_f \times \text{HHV} \times 3600$

$= \qquad \times \qquad \times 3600$

$=$ ______________ MJ/h

Energy to oil cooling, $Q_o = Q_g \times$ *factor*

$= \qquad \times$

$=$ ______________ kJ/h

Steam flow rate displaced, $f_{sd} = \dfrac{Q_o}{(1)}$

$=$ ______________

$=$ ______________ kg/h

Annual steam cost saving, $= f_{sd} \times h \times Cs$

$= \qquad \times \qquad \times$

= $______________/yr

Regenerator Installation On A Gas Turbine

Worksheet 14-T3
(Page 1 of 2)

Company: ____________________ Date: ____________________

Location: ____________________ By: ____________________

Turbine shaft power output, Wt_o ____________ kW

Initial fuel flow rate, f_{f1} ____________ L/s or m^3/s

Increase in turbine efficiency with regenerator (manufacturer) ____________ % (1)

Operation time, h ____________ h/yr

Fuel higher heat value, HHV (Appendix C) ____________ MJ/L or MJ/m^3

Unit fuel cost, Cf ____________ \$/L or $\$/m^3$

Initial energy input to the gas turbine, $Q_{g1} = f_{f1} \times \text{HHV} \times 3600$

$= \quad \times \quad \times 3600$

$=$ ____________ MJ/h

Initial gas turbine thermal efficiency, $Ef_{g1} = \dfrac{Wt_o \times 360}{Q_{g1}}$

$= \dfrac{\quad \times 360}{\quad}$

$=$ ____________ %

Revised gas turbine thermal efficiency, $Ef_{g2} = Ef_{g1} + (1)$

$= \quad +$

$=$ ____________ %

Regenerator Installation On A Gas Turbine

Worksheet 14-T3

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Revised fuel flow rate, $f_{f2} = f_{f1} \times \frac{Ef_{g1}}{Ef_{g2}}$

$= \quad \times \frac{}{\_\_\_\_}$

$=$ ____________ L/s or m^3/s

Annual fuel cost saving = $(f_{f1} - f_{f2}) \times h \times Cf \times 3600$

$= (\quad - \quad) \times \quad \times \quad \times 3600$

= $ ____________/yr

Improved Gas Turbine Performance

Worksheet 14-T4

(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

TURBINE DATA: (nameplate or measured)

Initial fuel flow rate, f_{f1} ____________ L/s or m^3/s

Revised fuel flow rate, f_{f2} ____________ L/s or m^3/s

Unit fuel cost, Cf ____________ \$/L or \$/m^3

Operation time, h ____________ h/yr

Annual fuel cost saving = $(f_{f1} - f_{f2}) \times h \times Cf \times 3600$

= (−   ) x   x   x 3600

= \$____________/yr

Steam Turbine As A Pressure Reducing Device

Worksheet 14-T5
(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Steam flow rate, f_s	______________ kg/h	
Inlet steam pressure	______________ kPa(absolute)	
Inlet steam enthalpy, h_i (h_g — Table 6)	______________ kJ/kg	
Outlet steam pressure	______________ kPa(absolute)	
Constant moisture per cent in outlet steam (Figure 24)	______________ %	(1)
Enthalpy of outlet steam, h_{go} (h_g — Table 6)	______________ kJ/kg	
Enthalpy of outlet condensate, h_{fo} (h_f — Table 6)	______________ kJ/kg	
Return condensate temperature	______________ °C	
Enthalpy of return condensate, h_r	______________ kJ/kg	
Alternator power output, Wa_o (Worksheet 14-E3 or manufacturer)	______________ kW	
Unit steam cost, Cs	$ ______________ /kg	
Unit electrical energy cost, Ce	$ ______________ /kWh	
Operation time, h	______________ h/yr	

Exhaust condensate flow rate, $f_{co} = f_s \times \frac{(1)}{100}$

$$= \qquad \times \frac{\qquad}{100}$$

= ______________ kg/h

Energy extracted from the steam,

$$Q_e = \frac{(f_s \times h_i) - (f_{co} \times h_{fo}) - [(f_s - f_{co}) \times h_{go}]}{1\ 000\ 000}$$

$$= \frac{(\qquad \times \qquad) - (\qquad \times \qquad) - [(\qquad - \qquad) \times \qquad]}{1\ 000\ 000}$$

= ______________ GJ/h

Steam Turbine As A Pressure Reducing Device

Worksheet 14-T5
(Page 2 of 2)

Company: ____________________ Date: ____________________

Location: ____________________ By: ____________________

Total steam system energy input, $Q_t = \dfrac{f_s \ \text{x} \ (h_i \ - \ h_r)}{1\ 000\ 000}$

$= \dfrac{\qquad \text{x} \ (\qquad - \qquad)}{1\ 000\ 000}$

$=$ ______________ GJ/h

Steam flow rate attributable to the turbine, $f_{st} = f_s \ \text{x} \ \dfrac{Q_e}{Q_t}$

$= \qquad \text{x} \ \dfrac{\qquad}{\qquad}$

$=$ ______________ kg/h

Note: This is the additional flow rate of inlet steam required to operate the turbine.

Annual steam cost $= f_{st} \ \text{x} \ h \ \text{x} \ Cs$

$= \qquad \text{x} \qquad \text{x}$

= $______________/yr (2)

Value of electrical energy produced $= Wa_o \ \text{x} \ Ce \ \text{x} \ h$

$= \qquad \text{x} \qquad \text{x}$

= $______________/yr (3)

Annual energy cost saving = (3) − (2) = −

= $ ______________/yr

Steam Turbine Inlet and Outlet Conditions

Worksheet 14-T6
(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Initial steam flow rate, f_{s1} ______________ kg/h

INLET CONDITIONS:

Initial inlet steam pressure, P_{i1} ______________ kPa(absolute)

Initial inlet steam temperature, T_{i1} ______________ °C

Initial enthalpy of inlet steam, h_{i1} (Figure 24) ______________ kJ/kg

Revised inlet steam pressure, P_{i2} ______________ kPa(absolute)

Revised inlet steam temperature, T_{i2} ______________ °C

Revised enthalpy of inlet steam, h_{i2} (Figure 24) ______________ kJ/kg

OUTLET CONDITIONS:

Initial outlet steam pressure, P_{o1} ______________ kPa(absolute)

Initial outlet steam temperature, T_{o1} ______________ °C

Initial enthalpy of outlet steam, h_{o1} (Figure 24 or Table 6) ______________ kJ/kg

Revised outlet steam pressure, P_{o2} ______________ kPa(absolute)

Revised outlet steam temperature, T_{o2} ______________ °C

Revised enthalpy of outlet steam, h_{o2} (Figure 24 or Table 6) ______________ kJ/kg

Operation time, h ______________ h/yr

Unit steam cost, Cs $ ______________ /kg

Steam Turbine Inlet and Outlet Conditions

Worksheet 14-T6

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Revised steam flow rate, $f_{s2} = f_{s1} \times \frac{h_{i1} - h_{o1}}{h_{i2} - h_{o2}}$

$= \quad \times \frac{\quad - \quad}{\quad - \quad}$

= ______________ kg/h

Annual steam cost saving = $(f_{s1} - f_{s2}) \times h \times Cs$

$= (\quad - \quad) \times \quad \times$

= $______________/yr

Electric Motor Drive Performance

Worksheet 14-E1
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Rated voltage, V_r ______________ volts

Rated full load current, I_r ______________ amps

Measured voltage, V ______________ volts

Measured current, I ______________ amps

Phase function, Y ______________
(1.73 for 3 phase, 2.0 for 2 phase, 1.0 for 1 phase)

Nameplate shaft power output ______________ kW

Rated full load power factor, $p.f._r$ ______________ (decimal)

Measured power factor, p.f. ______________ (decimal)

Unit electrical energy cost, Ce $ ______________ /kWh

Operation time, h ______________ h/yr

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r} = \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= ______________

Motor efficiency, Ef_m (Figure E-1) ______________ (decimal)

$$\text{Motor power input, } Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{\quad \times \quad \times \quad \times \quad}{1000}$$

= ______________ kW

Motor shaft power output, $Wm_o = Wm_i \times Ef_m$ = x = ______________ kW

Annual energy cost saving = $Wm_i \times h \times Ce$ = x x = $______________/yr

Motor Replacement

Worksheet 14-E2
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Motor Data

Initial motor power input, Wm_{i1} ______________ kW

Required motor shaft power output, Wm_o ______________ kW

Replacement motor rated shaft power output ______________ kW (1)

Operation time, h ______________ h/yr

Unit electrical energy cost, Ce $ ______________ /kWh

Load ratio $= \frac{Wm_o}{(1)}$

= ______________

= ______________

Replacement motor efficiency, Ef_m ______________ (decimal)
(Nameplate or Figure E-1)

Replacement motor power input, $Wm_{i2} = \frac{Wm_o}{Ef_m}$

= ______________

= ______________ kW

Annual energy cost savings = (Wm_{i1} − Wm_{i2}) x h x Ce

= (−   ) x   x

= $______________/yr

Electric Alternator Performance

Worksheet 14-E3
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Alternator Data: (nameplate or measured)

Rated voltage, V_r	______________ volts
Rated full load current, I_r	______________ amps
Measured voltage, V	______________ volts
Measured current, I	______________ amps
Phase function, Y (1.73 for 3 phase, 1.0 for 1 phase)	______________
Nameplate power output	______________ kW
Rated full load power factor, $p.f._r$	______________ (decimal)
Measured power factor, p.f.	______________ (decimal)

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

$$= \frac{\quad \times \quad \times \quad}{\quad \times \quad \times \quad}$$

= ______________

Alternator efficiency, Ef_a (Figure E-1) ______________ (decimal)

$$\text{Alternator power output, } Wa_o = \frac{V \times I \times Y \times p.f.}{1000}$$

$$= \frac{\quad \times \quad \times \quad \times \quad}{1000}$$

= ______________ kW

$$\text{Alternator shaft power input, } Wa_i = \frac{Wa_o}{Ef_a} = \text{\_\_\_\_\_\_\_\_\_\_} = \text{\_\_\_\_\_\_\_\_\_\_ kW}$$

Electric Motor Drives and Alternators

Most compressors are driven by alternating current electric induction motors, and many turbines drive alternators for electrical generation. Many small devices are driven by single phase motors, while the larger ones are driven by three phase motors. It is important to note that the number of phases affects electrical energy calculations.

Power Input to Motors

The power input to an electric motor can be expressed by the following equation.

$$Wm_i = \frac{V \times I \times Y \times p.f.}{1000}$$

Where, Wm_i = motor power input (kW)

V = measured voltage (volts)

I = measured current (amps)

Y = phase factor (1.73 for 3 phase, 2.0 for four wire 2 phase, 1.0 for single phase)

p.f. = measured power factor (decimal)

1000 = conversion from volt-amps to kilowatts.

Shaft Power Output From Motors

The mechanical power output of an electric motor is given by the folowing equation.

$$Wm_o = \frac{V \times I \times Y \times p.f. \times Ef_m}{1000}$$

Where, Wm_o = motor shaft power output (kW)

Ef_m = motor efficiency (decimal)

For all motors, the relationship between output and input is the same.

$Wm_o = Wm_i \times Ef_m$

Shaft Power Input To Alternators

The shaft power input to an alternator can be calculated by the following equation.

$$Wa_i = \frac{V \times I \times Y \times p.f.}{1000 \times Ef_a}$$

Where, Wa_i = alternator shaft power input (kW)

Ef_a = alternator efficiency (decimal)

The relationship between input and output is the same for all alternators.

$$Wa_i = \frac{Wa_o}{Ef_a}$$

Where, Wa_o = alternator power output (kW)

Power Output From Alternators

The power output from an alternator can be expressed by the following equation.

$$Wa_o = \frac{V \times I \times Y \times p.f.}{1000}$$

Where, V = measured voltage (volts)

I = measured current (amps)

Y = phase factor (1.73 for 3 phase, 1.0 for single phase)

$p.f.$ = measured power factor (decimal)

1000 = conversion from volt-amps to kilowatts

Obtaining Electrical Data

The data necessary to perform the calculations can be obtained by measurements and from electric motor and alternator performance data. Meters may be used to measure voltage, current and power factor.

Simplified performance data for power factor and motor efficiency is approximated by Figure E-1. Although the curves are representative of three phase induction motors, they may also be used for most other types of motors and for alternators. These curves should only be used in the absence of accurate measured or nameplate values.

The nameplate of a motor or an alternator is a reliable source of performance data. The Electrical Equipment Manufacturers Association requires that the nameplate provide the power rating of the motor or alternator in kilowatts or horsepower (labelled as kW or hp), the rated voltage, the full load current or amps (labelled as F.L. amps or F.L.A.), the power factor (labelled as P.F. or Pf) and the number of phases of the supply.

Load Ratio on a Motor or Alternator

The *load ratio* on a motor is the ratio of measured power input to the rated full load power input. The load ratio on an alternator is the ratio of measured power output to the rated full load power output. The load ratio for either a motor or an alternator can be expressed by the following equation.

$$\text{Load ratio} = \frac{I \times V \times p.f.}{I_r \times V_r \times p.f._r}$$

Where, I = measured current (amps)

V = measured voltage (volts)

$p.f.$ = measured power factor

I_r = rated full load current (amps)

V_r = rated voltage (volts)

$p.f._r$ = rated full load power factor

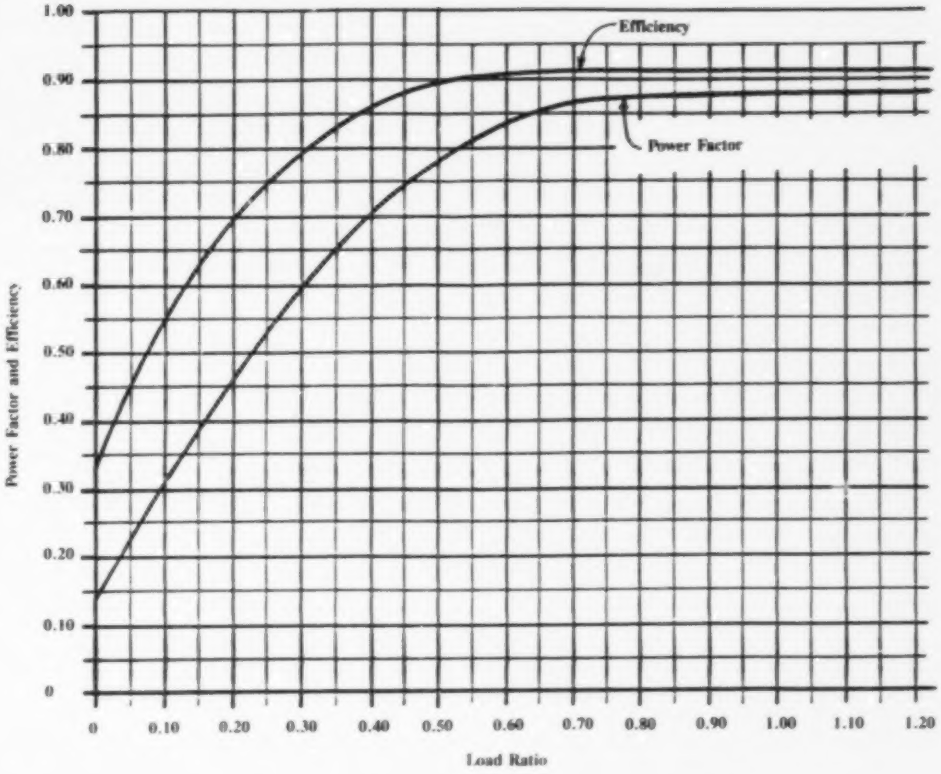
Using the calculated load ratio, the motor or alternator efficiency can be determined from Figure E-1. If the rated power factor is not known, a value from Figure E-1 can be used.

Cost of Electrical Energy

For the purposes of this module, the unit electrical energy cost is expressed as $/kWh consumed. This value is the incremental cost including the consumption rate, demand charges and any surcharges or discounts. The rate will vary for different electric power customers and for different regions. *Users of this module should establish an energy rate closely representative of their actual cost of electrical energy.*

Worksheets

Worksheets 14-E1, 14-E2 and 14-E3 are provided to assist with the analysis of motor or alternator performance.



Electric Motor Performance

Figure E-1